JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. II. FEBRUARY, 1890. No. 1.

The Society is not responsible for the opinions expressed by members, but merely puts them on record.

I.

MARSHALL VALVE GEAR.

By Passed Assistant Engineer Ira N. Hollis, U. S. Navy.

The Marshall valve gear is similar in principle to Hackworth's, whose invention probably gave rise to all the modern radial gears. Hackworth's consists essentially of a rod which obtains its motion at one end from an eccentric, while the other end is attached to a block which slides in a straight groove, or guide bar, inclined to a line from its center to the center of the shaft. Motion is communicated to the valve through a connecting rod attached to some point of the eccentric rod above mentioned. In Fig. 1, OE represents the eccentric, which is always placed exactly opposite the crank, ES the eccentric rod or lever, S the sliding block, FF_1 the guide bar, and VR the valve connecting rod attached to the point R of the eccentric lever.

As the crank revolves, the motion of R is the resultant of E in a circle and S in the straight line FF_1, whose inclination to OS may be changed by means of the quadrant QQ_1. When FF_1 coincides with OS, the motion of R is due entirely to the eccentric OE, and the travel of the valve is equal to 2 [lap + lead].

As the inclination of FF_1 is increased, the motion of R partakes more and more of that of S in the guide bar, and the opening of the steam port increases correspondingly, just as in

running the link from mid to full gear. The point of cut-off also depends upon the angle which FF_1 makes with OS.

This gear contains the essential features of all radial gears, an eccentric, or some equivalent which provides for the lap and lead, and which usually gives the valve a motion exactly opposite to that of the piston, and a guide bar, or equivalent, which regulates the maximum steam opening, the cut-off and the direction of motion of the engine.

In the figure, the crank would move in the direction of the hands of a watch, but if F were below OS the engine would be reversed. When the crank is on the center, the sliding block is always at S, near the center of FF_1, whatever the inclination of FF_1 may be. OS is at right angles with OH, and $ES = \sqrt{\overline{OE}^2 + \overline{OS}^2}$. The lead is therefore constant. As will be seen hereafter, the cut-off is varied exactly as in the link by varying the travel and wire-drawing more or less. This gear gives a very good distribution of steam, the motion of the valve being rapid at the point of cutting off, and slow during exhaust. There is also a fair range of cut-off without excessive wire-drawing. The wear of the sliding block and pin is very great, so great in fact as to preclude the general use of this gear, in spite of improvements to reduce the sliding friction.

Marshall has substituted for the straight guide bar, a swinging arm which has its center of oscillation in a line through S, at right angles to FF_1, and which describes an arc tangent to FF_1. The outer end of the eccentric lever, therefore, moves on an arc, which approximates more nearly to a straight line as the length of the swinging arm is increased. This modification, with the other parts exactly as Hackworth arranged them, constitutes the Marshall gear. Its simplest form is shown in Fig. 2, where the arc FF_1 described by the arm LS, oscillating about the pin L, takes the place of the guide bar. L may be raised or lowered by the arm QS and the cut-off changed, or the engine reversed, according to the inclination of FF_1 to SO. In the figure, the arm is set for going ahead. The point R may be located between O and S, or outside of S. In the latter case, the eccentric is usually keyed on the shaft in line with the crank, and a smaller eccentric

may be used. The lead is constant with this gear as with Hackworth's, and the distribution of steam is good, while the wear of the parts is not excessive. The travel is unequal on each side of the mid-position of the valve, and double ports are required on one end of the cylinder to obtain sufficient steam opening. A compensating rockshaft has been devised to correct this inequality, as shown in Fig. 3, which is a skeleton outline of the *Charleston's* valve gear.

In the figure, OE is the eccentric set with the crank, ER the eccentric lever, LT the radius link attached to the eccentric lever and swinging about the point L, RV the valve connecting rod, VN the short arm of the compensating rockshaft, V_1N the long arm, and V_1V_2 the valve link. The quadrant QAL swinging on the pin A is used for reversing and changing the cut-off by raising or lowering L. QQ_1 is connected with a steam reversing cylinder. The center of the reversing quadrant is situated on a line through O at right angles to the center line of the cylinder, and coincides exactly with T when the engine is on either center.

The motion of the valve may be obtained with fair accuracy from a modification of the Zeuner diagram. This modification, or correction, depends upon the parts of the gear, and affords a good method of comparing the ordinary link with the Marshall gear.

In Fig. 4, the parts are similarly lettered to those in Fig. 3, and the attachment of the valve connecting rod is taken outside of the radius link. OE_1 is the center of the cylinder, EE_1 the eccentric circle with the eccentric at OE, making the angle θ with OE_1. ETR is the eccentric lever, and FAF_1 the arc described by the radius link whose mid-position makes the angle α with the horizontal. The valve rod is attached at R. Let $OE = r$, $ET = l$, $TR = k$ and $LA = r_1$. Draw AT tangent to the arc FF_1 at A, and suppose, first, the radius link infinitely long, so that T moves on a straight line. T will be at a distance from the center line OA equal to $AT \sin \alpha = B_1T = AB_1 \tan \alpha$. Describe the arc TB with E as a center and ET as radius. For all ordinary cases, the distance BB_1, which depends upon the values of l and r, is exceedingly small and may be neglected. In the

figure, the parts are purposely chosen to exaggerate and show clearly BB_1 and the arc BT.

Let the angle $OBE = \varphi$. Then,

$$OB = l \cos \varphi + r \sin \theta. \tag{1}$$

and $$l \sin \varphi = r \cos \theta.$$

$$\cos \varphi = \sqrt{1 - \sin^2 \varphi} = \sqrt{1 - \frac{r^2}{l^2} \cos^2 \theta}.$$

Adding $\frac{r^4 \cos^4 \theta}{4 l^4}$, which is very small under the radical, we get

$$\cos \varphi = \sqrt{1 - \frac{r^2}{l^2} \cos^2 \theta + \frac{r^4}{4 l^4} \cos^4 \theta} = 1 - \frac{r^2}{2 l^2} \cos^2 \theta.$$

and equation (1) becomes,

$$OB = l - \frac{r^2}{2 l} \cos^2 \theta + r \sin \theta.$$

Since T must be at A when the crank is on either center, ET is necessarily equal to $E_1 A$, and

$$\overline{AO}^2 = \overline{E_1 T}^2 - \overline{E_1 O}^2 = l^2 - r^2.$$

$$AO = \sqrt{l^2 - r^2} = l \sqrt{1 - \frac{r^2}{l^2}} = l \left(1 - \frac{r^2}{2 l^2} \right) = l - \frac{r^2}{2 l} \text{ (nearly).}$$

The error here is exceedingly small.

$$AB = OB - OA = r \sin \theta + \frac{r^2}{2 l} \sin^2 \theta. \tag{2}$$

and

$$B_1 T = r \tan \alpha \sin \theta + \frac{r^2}{2 l} \tan \alpha \sin^2 \theta. \tag{3}$$

since BB_1 is practically very small.

The value of AB may be found graphically for any angle θ of the eccentric as follows: In Fig. 5 describe the circle OC with radius $= r$ and lay off $OO_1 = \frac{r^2}{2 l}$. With center O_1 describe a second circle with radius $= r$. Then, if OE be the eccentric, $AB = E_1 S_1 =$ the perpendicular from the point where OE prolonged cuts the second circle. This value will be found very accurate.

The proof of the correctness of this method is given, because it is similar to the graphic methods which follow.

Let the angle $O_1E_1O = \varphi_1$, then

$$OE_1 = O_1E_1 \cos \varphi_1 + OO_1 \sin \theta$$

$$= r \cos \varphi_1 + \frac{r^2}{2l} \sin \theta$$

and

$$r \sin \varphi_1 = \frac{r^2}{2l} \cos \theta$$

$$\cos \varphi_1 = \sqrt{1 - \sin^2 \varphi_1} = \sqrt{1 - \frac{r^2}{4l^2} \cos^2 \theta} = 1 - \frac{r^2}{8l^2} \cos^2 \theta.$$

The value of $\frac{r^2}{8l^2}$ is so small that $\cos \varphi_1$ is practically equal to unity. [If $\frac{r}{l} = \frac{1}{5}$, which it rarely exceeds, and $\theta = 0$, $\frac{r^2}{8l^2} \cos^2 \theta = \frac{1}{200}$.]

$$OE_1 = r + \frac{r^2}{2l} \sin \theta \quad \text{and,}$$

$$E_1S_1 = OE_1 \sin \theta = r \sin \theta + \frac{r^2}{2l} \sin^2 \theta$$

which is the value of AB given in Equation 2. The value of B_1T may be similarly constructed, as $\tan \alpha$ is a constant for the given position of AT.

[NOTE.—If greater accuracy is desired, when r is large compared with l, the value of AB may be taken equal to

$$r \sin \theta + \frac{r^2}{2l} (1 - \tan^2 \alpha) \sin^2 \theta$$

and OO_1 in the figure, laid off equal to

$$\frac{r^2}{2l} (1 - \tan^2 \alpha).]$$

To find the distance of R from the center line, we may consider the lever as subject to two independent motions, one of vibration about T and another of oscillation about E. If $\alpha = 0$ and T moves along the center line, the distance of R from OA produced will be $\frac{kr}{l} \cos \theta$.

If T moves along AT inclined to OA, and E along the line OA, the motion of R is due entirely to the oscillating motion about E, and its distance from the center line will be

$$\frac{l+k}{l} B_1T = \frac{l+k}{l} . r \tan \alpha \sin \theta + \frac{l+k}{l} . \frac{r^2}{2l} \tan \alpha \sin^2 \theta. \quad (5)$$

The actual distance of R from the center line, when E moves in a circle, and T along AT, is equal to the sum of these two quantities; so that,

$$\xi_1 = \frac{kr}{l} \cos \theta + \frac{l+k}{l} . r \tan \alpha \sin \theta + \frac{l+k}{2l^2} r^2 \tan \alpha \sin^2 \theta. \quad (6)$$

This value of ξ_1 is the distance of the valve from its central position. It applies to the Hackworth gear and is quite accurate. If the last term be neglected, it reduces to the equation of two circles, exactly equivalent to the Zenner diagrams for the ordinary eccentric and link. The construction of these two circles is shown in Fig. 6, where

$$Od = Od_1 = \frac{kr}{2l}$$

and

$$Dd = D_1d_1 = \frac{l+k}{2l} r \tan \alpha.$$

The two valve circles are described with centers at D and D_1 and give a fair approximation to the motion of the valve, especially when l is long compared with r, but they are exact only for the dead points of the eccentric and crank.

The value of

$$\frac{l+k}{2l^2} r^2 \tan \alpha \sin^2 \theta$$

may be obtained graphically and laid off as a correction on the circles. In the figure, make

$$OO_1 = \frac{k+l}{4l^2} r^2 \tan \alpha.$$

Describe the circle OC with any convenient radius [it should not be less than 5 or 6 times OO_1] and the second circle O_1C with center at O_1. Then, for any angle θ, the value of

$$\frac{k+l}{4l^2} r^2 \tan \alpha \sin^2 \theta$$

is equal to the intercept between the two circles OC and O_1C at twice that angle.

Its value for the angle COC_1 equals ef. This correction may be laid off from the valve circle along the radius OC_1, and by obtaining its value for a sufficient number of crank angles, the whole valve curve may be drawn in as shown in the figure. It will be noticed that this correction is laid off positively on the circle D and negatively on D_1, while its numerical value is exactly the same for all points 180° apart. The valve curves show plainly the difference of travel on each side of the central position and the difference of steam opening for the two ends of the cylinder with equal leads.

This inequality is due to the fact that T travels farther on the upper side of A than on the lower. It decreases rapidly as the ratio of l and r increases.

In the case taken, which is extreme, the cut-offs are very unequal for the two ends of the cylinder; but if T were on the right-hand side of OR_1 and AT inclined the other way, the correction $\frac{k+l}{4l^2}\, r^2 \tan \alpha \sin^2\theta$ would partly balance the inequalities of cut-off due to angularity of connecting rod. In fact, l and r may be so chosen as to compensate exactly for the connecting rod, and the cut-offs made exactly the same on the two ends, but at the sacrifice of steam opening on one end.

The above applies to the Hackworth gear, which is seldom used, and it is given only because it has direct bearing on Marshall's. If R is located between E and T, the same formulæ apply making k negative in the second and third term of Equation 7; so that

$$\xi_1 = \frac{kr\cos\theta}{l} + \frac{l-k}{l}\, r \tan\alpha \sin\theta + \frac{l-k}{2l^2}\, r^2 \tan\alpha \sin^2\theta. \quad (7)$$

The same valve circles are approximately correct for the Marshall gear with a finite radius link, and the corrections for errors due to the arc FF_1 are easily found.

If T moved along the arc FF_1 instead of the straight line, it would be found at a point between T_1 and T_2 for the eccentric angle θ. Supposing it to be at T_1 in the line drawn through B

at right angles to OR, which introduces only a slight error into the result, we have

$$DB = DA - AB$$

$$DA = r_1 \sin \alpha \text{ (call its value } q)$$

and $$AB = r \sin \theta + \frac{r^2}{2l} \sin^2\theta \text{ [see Equation 2]}$$

$$DB = q - r \sin \theta - \frac{r^2}{2l} \sin^2\theta$$

$$\overline{D_1T_1}^2 = r_1^2 - \overline{DB}^2$$

$$= r_1^2 - \left(q - r \sin \theta - \frac{r^2}{2l} \sin^2\theta \right)^2$$

$$D_1T_1 = \sqrt{r_1^2 - q^2 - r^2 \sin^2\theta - \frac{r^4}{4l^2} \sin^4\theta + 2qr \sin \theta + \frac{qr^2}{l} \sin^2\theta + \frac{r^3}{l} \sin^3\theta}.$$

In this equation, $r_1^2 - q^2 = q_1^2$, and

$$D_1T_1 = q_1 \sqrt{1 - \frac{r^2}{q_1^2} \sin^2\theta - \frac{r^4}{4q_1^2l^2} \sin^4\theta + \frac{2qr}{q_1^2} \sin \theta + \frac{qr^2}{lq_1^2} \sin^2\theta + \frac{r^3}{lq_1^2} \sin^3\theta}.$$

The trigonometric quantities under the radical form only a small fraction of the value of D_1T_1 and the terms containing $\sin^3\theta$ and $\sin^4\theta$ will be found so small that they may be neglected.

The equation, therefore, becomes approximately

$$D_1T_1 = q_1 \left(1 + \frac{qr}{q_1^2} \sin \theta - \frac{r^2}{2q_1^2} \sin^2\theta + \frac{qr^2}{2lq_1^2} \sin^2\theta \right)$$

$$= q_1 + \frac{qr}{q_1} \sin \theta - \frac{r^2}{2q^1} \sin^2\theta + \frac{qr^2}{2lq_1} \sin^2\theta \qquad (8)$$

$$BT_1 = D_1T_1 - D_1B = D_1T_1 - q_1$$

$$= \frac{qr}{q_1} \sin \theta - \frac{r^2}{2q_1} \sin^2\theta + \frac{qr^2}{2lq_1} \sin^2\theta.$$

As may readily be seen $\frac{q}{q_1} = \tan \alpha$ and

$$BT_1 = r \tan \alpha \sin \theta + \frac{r^2}{2l} \tan \alpha \sin 2\theta - \frac{r^2}{2q_1} \sin 2\theta \qquad (9)$$

and the value of

$$\xi_1 = \frac{kr}{l} \cos \theta + \frac{l+k}{l} BT_1$$

$$\xi_1 = \frac{kr}{l} \cos \theta + \frac{l+k}{l} r \tan \alpha \sin \theta + \frac{l+k}{2l^2} r^2 \tan \alpha \sin 2\theta$$

$$- \frac{l+k}{2l} \times \frac{r^2}{q_1} \sin 2\theta. \qquad (10)$$

This is precisely the same as that given in Equation 6 for Hackworth's gear, excepting the last term, which forms the correction due to a finite radius link.

$\frac{r^2}{2q_1} \sin 2\theta$ is approximately equal to the intercept of D_1T_1 between FF_1 and AT. Neglecting the terms in $\sin 2\theta$, Equation 10 becomes

$$\xi_1 = \frac{kr}{l} \cos \theta + \frac{l+k}{l} r \tan \alpha \sin \theta. \qquad (11)$$

The valve circles represented by this equation may be constructed as in Fig. 6. In Fig. 13, D and D_3 are the centers of the two valve circles $a - a - a$ and $a_1 - a_1 - a_1$, obtained with

$$Od = Od_1 = \frac{kr}{2l} \qquad \text{and} \qquad Dd = D_3d_1 = \frac{l+k}{2l} r \tan \alpha.$$

After the valve circles are drawn in, the correction due to

$$\frac{l+k}{2l^2} r^2 \tan \alpha \sin 2\theta - \frac{l+k}{2l} \cdot \frac{r^2}{q_1} \sin 2\theta$$

of Equation 10 may be applied exactly as stated before. Lay off

$$OO_1 = \frac{l+k}{4l^2} r^2 \tan \alpha - \frac{(l+k)r^2}{4lq_1},$$

and describe the circles OH and O_1H as previously in Fig. 6.

The correction for any angle is found in the intercept between these two circles for twice the angle. For instance, the correction on OY is H_1H_2. In the Marshall gear

$$\frac{l+k}{2l^2} r^2 \tan \alpha - \frac{l+k}{4l} \cdot \frac{r^2}{q_1}$$

will usually be negative, and the correction must be laid off

inside the circles between 0° and 180°, and positive between 180° and 360°.

There is a slight error in the above, due to the angularity of the valve connecting rod. In Fig. 7, let RV be the valve connecting rod of length l_1. The distance of the point R above its central position VS will be approximately $r \sin \theta$ for an eccentric angle θ. The projection of RV on the center line of the valve stem will be VS, such that

$$VS = \sqrt{\overline{RV}^2 - \overline{RS}^2} = \sqrt{l_1^2 - r^2 \sin^2\theta} = l_1 - \frac{r^2}{2l_1} \sin^2\theta$$

and $SS_1 = \dfrac{r^2}{2l_1} \sin^2\theta$, which is the error sought for. It may be added to Equation 11, giving for the distance traveled by the valve from its central position—

$$\xi_1 = \frac{kr}{l} \cos\theta + \frac{l+k}{l} r \tan\alpha \sin\theta + \frac{l+k}{2l^2} r^2 \tan\alpha \sin^2\theta + \frac{r^2}{2l_1} \sin^2\theta - \frac{l+k}{2l} \cdot \frac{r^2}{q_1} \sin^2\theta.$$

Then in Figure 13, OO_1 should be laid off equal to

$$\frac{l+k}{4l^2} r^2 \tan\alpha + \frac{r^2}{4l_1} \sin^2\theta - \frac{(l+k) r^2}{4l q_1},$$

and the corrections obtained as before.

The same thing in regard to cut-off and inequality of steam opening as stated for the Hackworth gear, is also true for the Marshall gear.

The dotted circles $a - a - a$ and $a_1 - a_1 - a_1$, become the curves $b - b - b$ and $b_1 - b_1 - b_1$ by applying the corrections, and the difference of travel on the two sides of the center may be seen at a glance. Drawing the lap circles SS_2 and S_1S_3, we have the cut-off at the crank positions OC_4 and OC_8, which correspond to exactly half stroke for both ends of the cylinder.

The maximum opening of the ports are not equal. They differ by twice the intercept between the circles OH and O_1H for twice the angle *Dod*, if the lap is the same on both ends.

In this case, and all following, the cylinder is supposed to lie to the right of O, and the piston to be on its outboard center for

the crank position OX. The eccentric also coincides with the crank. If the eccentric were opposite to the crank, as it would be with R between E and T (see Fig. 4), the same curves would hold good with the cylinder to the left of O and the outboard center to the left; k would also be negative in all terms containing $(l + k)$.

In this figure the crank positions for steam opening are shown at OC_2 and OC_6.

The influence of the different parts of the gear is shown by the formula

$$\xi_1 = \frac{kr}{l}\cos\theta + \frac{l \pm k}{l} r \tan\alpha \sin\theta = \frac{l \pm k}{2l^2} r^2 \tan\alpha \sin 2\theta + \frac{r^2}{2l_1}\sin^2\theta - \frac{l \pm k}{2lq_1} r^2 \sin^2\theta. \quad (12)$$

The first term $\frac{kr}{l}\cos\theta$ is dependent upon k, r and l, which must have such value that $\frac{kr}{l}$ = 2 lap + 2 lead.

This is true, whether the eccentric be opposite the crank or with it, and whether the valve connecting rod be attached inside of the radius link or outside.

Since k must be less than l for the inside connection, the ratio $\frac{k}{l}$ cannot be made so large as for the outside, where there is no limit to k; the eccentricity must, therefore, be greater in the former case, to provide for the same lap and lead.

The term $\frac{l \pm k}{l} r \tan\alpha$ determines the opening of the port, which, therefore, depends upon the eccentricity, the ratio $\frac{l \pm k}{l}$ and the tangent of the angle which the radius link makes with the horizontal. Hence, the greater the angle α for fixed values of k, l and r, the greater the port opening.

The lead must remain constant for a given lap, as the valve circle will always have its center on the line Dd parallel to OY and at a distance $\frac{kr}{2l}$ from it.

The last three terms form the correction for the departure from valve circles, and by means of these terms the points of cut-off may be equalized.

The term

$$\frac{l \pm k}{2l^2} r^2 \tan a \sin 2\theta$$

is additive in the first quadrant. It increases rapidly with the increase of the eccentricity, or the decrease of the eccentric lever.

The term $\frac{r^2}{2l_1} \sin 2\theta$ is usually small, depending on the valve-connecting rod, which is long compared with r.

The last term

$$\frac{l \pm k}{2lq_1} r^2 \sin 2\theta$$

is negative. It is so large, compared with the two terms just mentioned, that the whole correction is negative for the first and second quadrants.

The equalization of cut-off depends upon

$$\frac{l \pm k}{2lq_1} r^2 \sin \theta,$$

and, as may be seen, directly upon q_1, if k, l and r be fixed.

Now $q_1 = r_1 \cos a$. The value of this term, therefore, increases as the radius link decreases, or *vice versa;* so that for a given valve gear, the point of cut-off may be materially changed by the radius link without disturbing any other parts.

Such values can be given to k, l and r, that

$$\frac{l \pm k}{2l^2} r^2 \tan a + \frac{r^2}{2l_1} = \frac{l \pm k}{2lq_1} r^2,$$

and the correction disappears for a given point of cut-off, thus making the Marshall valve gear practically equivalent to the eccentric and link.

In this case the maximum steam opening and lead are the same for both ends, but the cut-offs vary, and the parts would not work very smoothly together. The equality of opening with the same cut-offs at both ends may easily be secured with double ports at one end of the cylinder, as stated before.

The rockshaft, with arms set at different angles, as shown in Fig. 3, is introduced to equalize the travel on each side of the central position, without interfering with the cut-off.

Let NV_1 in Fig. 8, be connected with the valve stem, and NV with the eccentric lever, and let the angle $VNV_1 = \beta$.

Take $NV_1 = m$ and $NV = n$. As the valve connecting rod moves practically parallel to itself, the distance moved by V for a given travel ξ_1 of the eccentric lever pin would be VV_2.

Take $VNV_2 = \varphi = V_1NV_3$; then

$$\xi_1 = V_2v_1 - Vv = n \sin(\beta - \varphi) - n \sin \beta$$

or

$$\xi_1 + n \sin \beta = n \sin \beta \cos \varphi + n \cos \beta \sin \varphi \qquad (13)$$

which is the equation of a circle whose center has the co-ordinates $\frac{n \sin \beta}{2}$ and $\frac{n \cos \beta}{2}$

If this circle be drawn, the value of φ may be found graphically for any valve of $\xi_1 + n \sin \beta$.

In Fig. 9, lay off $ON = \frac{n \sin \beta}{2}$ and $Nm_1 = \frac{n \cos \beta}{2}$ and draw the circle $m_1 o$ with m_1 as center.

The angle φ is obtained for any value of ξ_1 by laying off $\xi_1 + n \sin \beta = Oa$; $\varphi = aOX$.

The distance moved by the valve from its central position $= \xi = m \sin \varphi$.

Take

$$\rho = \xi_1 + n \sin \beta = n \sin \beta \cos \varphi + n \cos \beta \sin \varphi \qquad (14)$$

and

$$\rho_1 = n \sin \beta \cos \varphi + (n \cos \beta - m) \sin \varphi. \qquad (15)$$

Then

$$\rho - \rho_1 = m \sin \varphi = \xi.$$

The circle of equation 15 may be drawn and the value of ξ determined graphically at once. Lay off $m_1n_1 = \frac{m}{2}$ and draw the second circle n_1O; then, the value of ξ is equal to the intercept between the two circles for any angle φ. When $\varphi = aOX$, $\xi = ab$.

This affords an accurate method of obtaining φ which can be employed when the parts of the valve gear are all determined, but it is tedious in designing a new gear. The following is sufficiently accurate for all practical purposes:

From Equation 13,

$$\xi_1 = n \sin \beta \cos \varphi + n \cos \beta \sin \varphi - n \sin \beta.$$

and

$$\frac{m\xi_1}{n \cos \beta} = m \tan \beta \cos \varphi + m \sin \varphi - m \tan \beta.$$

$$= m \tan \beta \sqrt{1 - \sin^2 \varphi} + m \sin \varphi - m \tan \beta$$

$$= m \tan \beta - \frac{m \tan \beta}{2} \sin^2 \varphi + m \sin \varphi - m \tan \beta \text{ (nearly)}$$

$$= m \sin \varphi - \frac{m \tan \beta}{2} \sin^2 \varphi$$

$$= \xi - \frac{m \tan \beta}{2} \sin^2 \varphi$$

and

$$\xi = \frac{m\xi_1}{n \cos \beta} + \frac{m \tan \beta}{2} \sin^2 \varphi. \qquad (16)$$

Since $\xi = m \sin \varphi$, $\quad \frac{\xi^2}{m} = m \sin^2 \varphi$

and (16) becomes

$$\xi = \frac{m\xi_1}{n \cos \beta} + \frac{\tan \beta}{2m} \xi^2. \qquad (17)$$

This amounts to multiplying all the terms of Equation 12 by

$$\frac{m}{n \cos \beta} \text{ and adding } \frac{\tan \beta}{2m} \xi^2,$$

so that,

$$\xi = \frac{m}{n \cos \beta} \cdot \frac{kr}{l} \cos \theta + \frac{m}{n \cos \beta} \cdot \frac{l + k}{l} r \tan a \sin \theta$$

$$+ \frac{m}{n \cos \beta} \cdot \frac{l + k}{2l^2} r^2 \tan a \sin^2 \theta + \frac{m}{n \cos \beta} \cdot \frac{r^2}{2l_1} \sin^2 \theta$$

$$- \frac{m}{n \cos \beta} \cdot \frac{(l + k) r^2}{2q_1 l} \sin^2 \theta + \frac{\tan \beta}{2m} \xi^2 \qquad (18)$$

This apparently complex equation is very easily solved graphically.

In Fig. 10, lay off

$$Od = Od_1 = \frac{m}{n \cos \beta} \cdot \frac{kr}{2l}$$

and

$$Dd = D_1 d_1 = \frac{m}{n \cos \beta} \cdot \frac{l + k}{2l} \cdot r \tan a$$

and construct the two valve circles with centers D and D_1.

Then lay off

$$OO_1 = \frac{m}{n \cos \beta} \left(\frac{l + k}{4l^2} r \tan a + \frac{r^2}{4l_1} - \frac{(l + k) r^2}{4q_1 l} \right) \quad (19)$$

and draw the circles OH and O_1H.

The circle DO is very nearly correct for the motion of the valve and OE may be taken as equal to the maximum travel of the valve on one side of its center.

In the figure, lay off OC, so that $HOC = 2\ HOE$ and draw O_1H_3 parallel to OC.

Then to obtain the value of $\frac{\tan \beta}{2m} \xi^2$ for any eccentric angle, lay off $O_1O_2 = \frac{\tan \beta}{4m}\ \overline{OE}^2$, and describe the circle O_2H_2 with O_2 as center, H_2 being the point where the circle O_1H cuts the line H_3O_1 produced. [NOTE.—The distance OH may be taken at any convenient length, so long as it exceeds OO_1 five or six times.]

The correction on the valve circle for any angle $HOC_2 = \theta$ is composed of two parts, one the intercept between the circles OH_1 and O_1H_4 at the angle $HOC_3 = 2\theta$, and the other the intercept between O_1H_4 and O_2H_2 at the same angle. The difference between these two is the intercept between the circles O_2H_2 and OH_1. Hence the correction for any angle θ is found in the intercept between the circles O_2H_2 and OH_1 at twice that angle.

This is positive in the first quadrant and negative in the third when measured outside of the circle OHH_1, and *vice versa*.

It changes sign whenever the auxiliary circles cross each other.

Fig. 12 represents the cross sections applied to both valve circles. The two circles OH_1 and O_2H_2 give the required intercept

as in Fig. 10. The dotted circles a—a and a_1—a_1 with D and D_1 as centers, represent the approximate valve circles, and the full curves b—b and b_1—b_1 the actual valve curves.

As shown in these diagrams, the lead is increased on one end by t_1t and decreased on the other by the same amount. Drawing the lap circles s and s_1 it will be seen that the port openings and points of cut-off are about the same for both ends, while the lead varies slightly. It is neither constant nor the same on both ends, but is greater on the outboard end where the piston has the greatest initial velocity.

All the terms of Equation 18 are similar to those in Equation 12, excepting the last term, and the same remarks may be made as to their functions. The last term $\frac{\tan \beta}{2m} \xi^2$, is entirely corrective of irregularities in the simpler gear, and depends for its value upon the arms and angle of the rockshaft. The value of

$$\frac{m}{n \cos \beta} \left(\frac{l + k}{4l^2} r^2 \tan \alpha + \frac{r^2}{4l_1} - \frac{(l + k) r^2}{4q_1 l} \right)$$

is negative and by giving m, n, and β such values that $\frac{\tan \beta}{2m} \xi^2$ is numerically equal to it, Equation 18 becomes nearly a circle and the gear is equivalent to eccentrics and link.

The quantity $\frac{m}{n \cos \beta}$ must vary to suit the particular valve gear, but n must be at least equal to the total maximum travel of the valve, and β must not exceed 30°; otherwise the connections will be crank and the strains irregular.

Before discussing the question of design, it may be well to refer to Fig. 11, which represents the complete diagram for the *Charleston's* engine at $\frac{7}{10}$ cut-off. The sketch of Fig. 3 is taken from this engine.

The first method of obtaining ξ is employed here, because it shows clearly the effect of the rockshaft upon the valve motion, although the second method is shorter and accurate enough.

The parts taken from the engine, as actually constructed, have the following values: $l = 23.''1$, $r = 2.5$, $OA = 23.''$, $k = 16.''65$, $l + k = 39.''75$, $r_1 = 16.''5$, $\alpha = 25°50'$ and $\tan \alpha = .484$,

$q^1 = 14.''85$, $l_1 = 74\frac{5}{8}$, $m = 11''$, $n = 8\frac{3}{4}''$ and $\beta = 24°9'$. Then

$$Od_1 = \frac{kr}{2l} = 0.''9$$

and

$$D_1d_1 = \frac{(l+k)\ r \tan a}{2l} = 1''.04.$$

The circle D_1O is drawn half size, and the opposite valve circle likewise.

$$\frac{l+k}{4l^2}r^2 \tan a + \frac{r^2}{4l_1} - \frac{l+k}{4q_1l}r^2 = 0.''105,$$

and the corrections on the valve circles are found just as shown in Figs. 6 and 13.

The actual work of construction is left out to avoid confusion. The full lines b–b represent the values of ξ_1 or the travel due to the extreme end of the eccentric lever without compensating rockshaft.

$$\frac{n \sin \beta}{2} = 1.''79, \text{ and } \frac{n \cos \beta}{2} = 3.''99.$$

The points m_1 on each side of OY are laid off from these.

Draw the circles ON_1T and ON_2T_1 with m_1 as centers, and lay off $m_1n_1 = \frac{m}{2}$ parallel to OY.

Describe the second circles ONT and ON_2T_1 and the complete circle OT. Radii are drawn for every 15° in order to be quite accurate, but every 30° is enough ordinarily.

By adding the radius of the circle OT to every value of ξ_1 and describing from O short arcs cutting the circle ON_1T, and joining the points thus obtained with O, we get the values of ξ. They are numbered on the circle beginning with OX as 1.

The values of ξ corresponding to the circle D_2 are drawn to the left of Y. The full lines $b_1 - b_1$ represent the curves obtained. By drawing the lap circles with $OS = 1\frac{1}{16}''$, and $OS_1 = 1\frac{1}{2}''$ as taken from the drawings, we can measure all the important points of steam distribution.

The following table represents the results as measured on the diagram and on the engine after it was set up in the ship:

	From Engine.		From Diagram.	
	Outbd.	Inbd.	Outbd.	Inbd.
Cut-off in fraction of stroke, .	.712	.688	.716	.688
Lead in inches, ...	$\frac{31}{32}$	$\frac{27}{32}$	$\frac{15}{16}$	$\frac{7}{8}$
Travel in inches, ...	$7\frac{1}{2}$		$7\frac{9}{16}$	
Maximum steam opening, .	$2\frac{3}{16}$	$2\frac{7}{32}$	$2\frac{1}{4}$	$2\frac{3}{16}$

The approximate circles are dotted in from the centers D and D_3. In the above the question of exhaust is left out, as it is only a matter of drawing in the exhaust lap.

The design of this valve gear is comparatively easy with the aid of the above curves. The dimensions of some of the parts are fixed approximately by the engine. It is desirable to make the eccentricity as small as possible to reduce inequalities of distribution in the two ends. This can readily be accomplished where the valve rod is attached outside of the radius link. The length of l depends somewhat on the framing of the engine and the location of the reversing gear, while the value of $l + k$ should not greatly exceed the distance between the center lines of shafting and valve stem.

The valve connecting rod and the arms of the compensating rockshaft are fixed by the design of the engine. The angle β and the length of the radius link must be determined from the diagrams, to satisfy conditions required.

This gear cannot be made to cut off at the same point on the two ends for all grades of expansion. It can be adjusted for some given cut-off, and will then be approximately correct for others. Usually it is well to select some ratio of expansion as the normal ratio for the engine, and that at which all irregularities should be gotten rid of in the design. It need not necessarily be for the maximum power.

Take a case of the simple gear, and suppose the cut-off to be at midstroke, the maximum opening of the steam port to be $1\frac{1}{2}''$ and the valve to begin opening when the piston is .01 of its stroke from the end.

First, consider the connecting rod infinitely long, and draw valve circles to satisfy the above conditions. In Fig. 13, OY will be the position of the crank for steam closing, OC_2 for steam opening, D and D_3 the centers of the valve circles to give maximum opening of $1\frac{1}{2}''$, and SS_2 and S_1S_3 the lap circles. The construction of these circles is taken directly from Zeuner. The equation of the valve circles will be

$$\xi_1 = \frac{kr}{l}\cos\theta + \frac{l+k}{l} r \tan a \sin\theta$$

which becomes

$$\xi_1 = \frac{l+k}{l} r \tan a,$$

when $\theta = 90°$ or $270°$. The lap is therefore just equal to

$$\frac{l+k}{l} r \tan a.$$

The connecting rod cannot be infinitely long, and there would be considerable inequality of cut-off on the two ends, if OY were the crank position corresponding to the cut-off. Taking the connecting rod at four times the crank, OC_4 and OC_8 will be the two crank positions for $\frac{1}{2}$ cut-off, if the outboard end of the cylinder be to the right. Therefore the correction on the valve circle a—a must be such that the valve curve passes through the intersection of SS_2 and OC_4. The same applies to the circle a_1—a_1, or as may readily be seen, the same amount is taken from a—a at OC_4 that is added to a_1—a_1 at OC_8.

Let r_2 = length of crank, and l_2 = length of connecting rod. Then, when the piston is at midstroke,

$$\cos\theta = \frac{r_2}{2l_2}, \text{ and } \sin\theta = 1 \text{ (nearly).}$$

Substituting in Equation 12, we have—

$$\xi_2 = \frac{kr}{l}\cdot\frac{r_2}{2l_2} + \frac{l+k}{l} r \tan a + \frac{l+k}{2l^2} r^2 \tan a + \frac{r^2}{2l_1} - \frac{l+k}{l}\cdot\frac{r^2}{2q_1} \text{ on } OC_4$$

$$\xi_3 = \frac{kr}{l}\cdot\frac{r_2}{2l_2} - \frac{l+k}{l} r \tan a + \frac{l+k}{2l^2} r^2 \tan a + \frac{r^2}{2l_1} - \frac{l+k}{l}\cdot\frac{r^2}{2q_1} \text{ on } OC_8$$

Since ξ_2 and ξ_3 must be equal to the lap, if cut-off takes place, and since the lap $= \frac{l+k}{l} r \tan a$, we must have

$$\frac{kr}{l} \cdot \frac{r_2}{2l_2} + \frac{l+k}{2l^2} r \tan a + \frac{r^2}{2l_1} - \frac{l+k}{l} \cdot \frac{r^2}{2q_1} = 0 \quad (21)$$

From the construction of the circles,

$$\frac{kr}{2l} = Od. \quad (22)$$

$$\frac{l+k}{2l} r \tan a = Dd. \quad (23)$$

[These equations are not deduced for this case only, as they apply to any gear for half cut-off. Other equations may be obtained for other points of cut-off, but they are not necessary in the great majority of cases.]

The solution of the three equations above will give such values of k, r, l, q, &c., as will satisfy the requirements of the case. Od and Dd are measured on the diagram as constructed for a connecting rod of infinite length. The lap and lead are already fixed. In the special case under consideration, $l + k$ being fixed by the distance between center lines of cylinder and valve stem, may be assumed; call it $= 40''$; for similar reasons $l_1 = 70''$.

From the diagram

$$Od = 1.''45 \quad \text{and} \quad Dd = 1.01$$

$\frac{r_2}{2l_2} = \frac{1}{8}$ since the connecting-rod is four times the crank. Equations 21, 22 and 23 become

$$\frac{2.90}{8} + \frac{40}{4l^2} \cdot r^2 \tan a + \frac{r^2}{140} - \frac{40}{l} \cdot \frac{r^2}{2q_1} = 0$$

$$\frac{kr}{2l} = 1.45 \quad . \quad \frac{l+k}{2l} r \tan a = 1.01$$

$$\text{and } l + k = 40.$$

In these equations there are five unknown quantities, any one of which may be assumed and the others worked out. The

results will satisfy the condition of $\frac{1}{2}$ cut-off, whatever value is assumed for one part, but it may require a few trials to obtain values which will give a good gear for other points of cut-off. A little judgment is required, with some experience with the gear at work, to obtain the best results.

Suppose $r = 2.''5$, then $k = 21.''5$, $l = 18.''5$; $\tan a = .374$, $a = 20°30'$ and $q_1 = 12.5$. Also $\frac{q_1}{\cos a} = r_1 = 13.''34$.

Substituting back, the correction circles may be drawn exactly as before and the valve curves determined. The value of a may be varied and curves for other points of cut-off be found. In the figure, the circles and curves D_1 and D_4 are drawn to show the points of cut-off for crank angles OC_3 and OC_7, .34 and .38 of the stroke respectively. The maximum port openings are modified, but not seriously. The exhaust may be examined on the same curve. With double ports at one end a very good distribution of steam may be obtained. The proportions of the compensating rockshaft may also be obtained in the above manner.

Take the mean cut-off at $\frac{1}{2}$ stroke, the maximum steam opening at $1\frac{1}{2}''$, and let the port opening take place when the piston is .015 of its stroke from the end. The valve circles are constructed as if the connecting rod were infinitely long and the lap circles S_2S_3 and S_4S_5 are drawn in, precisely the same as before. See Fig. 12. The equation of these circles will be

$$\xi = \frac{m}{n\cos\beta}\cdot\frac{kr}{l}\cos\theta + \frac{m}{n\cos\beta}\cdot\frac{l+k}{l}r\tan a\sin\theta$$

from Equation 18.

Measuring on the drawing, we have

$$\frac{mkr}{nl\cos\beta} = 2Od = Ot_1 = 2.''8$$

and

$$\frac{mr(l+k)\tan a}{nl\cos\beta} = 2Dd = 1.''85.$$

When $\theta = 90°$ or $270°$,

$$\xi = \frac{mr(l+k)\tan a}{nl\cos\beta}$$

which is the lap for the $\frac{1}{2}$ cut-off.

If the connecting rod be finite, say 4 times the crank, the cut-off must take place at the crank positions OC_3 and OC_5 as before; where $\cos\theta = \frac{r_2}{2l_2}$ and $\sin\theta = 1$ (nearly). Suppose β in Equation 18 be equal to 0, and the valve gear simple, with rockshaft arms in line; then, the correction for $\frac{1}{2}$ cut-off would be v_1v_2 on OC_3, which is practically equal to the correction at 90°. Therefore, lay off $OO_1 = \frac{1}{2}v_1v_2$ and describe the two circles OH and O_1H. The intercepts between these two circles will be the corrections on the valve circles as before. Draw OC_6 so that $HOC_6 = 2\,HOE$ and O_1H_6 parallel to it.

The travel of the valve is less than OE by the intercept H_3h_3 between the two circles, and it is this decrease of travel or steam opening which must be restored by setting the rockshaft arms at an angle β. In making this compensation, it will be found that the lead will be increased somewhat, for which a second correction must be applied—large enough to allow the lap to be increased slightly.

The value of β can be worked out analytically, but the process is tedious and unnecessary. A few trials in locating the point O_2, will give the third circle O_2H_2 such that the new corrections will satisfy all requirements.

As a help, lay off OC_7 at twice the angle OC_3 from OX. Then the third auxiliary circle should make the intercepts H_3h_3 and h_1h_2 about equal and slightly smaller than Hh. H_3h_3 represents the increase in the travel over the circle, and therefore the increase in the lap. A little reflection will show that h_1h_2 also represents a decrease in the correction on OC_3, since it represents an increase of the lap, in order to preserve the cut-off at $\frac{1}{2}$ stroke. The increase of lead Hh cannot be entirely counterbalanced without reducing the greatest steam opening. With the two circles OHH_1 and $O_2H_3H_2$, the new valve curve may be constructed and the parts of the valve gear obtained. [All that has been said of the circle DO and its corrections applies equally to the circle D_1O.]

If $\theta = 0$ in Equation 18,

$$\xi = \frac{m}{n\cos\beta}\cdot\frac{kr}{l} + \frac{\tan\beta}{2m}\,\xi^2 = Ot$$

where $$\frac{m}{n\cos\beta}\cdot\frac{kr}{l}=Ot_1$$

and $$\frac{\tan\beta}{2m}\cdot\xi^2=t_1t.$$

ξ may be taken equal to Ot_1 for all practical purposes, and

$$\frac{\tan\beta}{2m}(Ot_1)^2=t_1t.$$

In the case taken, let $l+k=40''$ and $l_1=70$. We have, then, the four equations—

$$l+k=40\text{ (a)},\qquad \frac{mkr}{nl\cos\beta}=2.8\text{ (b)},$$

$$\frac{mr(l+k)\tan\alpha}{nl\cos\beta}=1.85\text{ (c)},\quad\text{and}\quad\frac{\tan\beta}{2m}(Ot)^2=t_1t=.125\text{ (d)}$$

to find the parts of the gear.

Some of these parts may be assumed and the others worked out to correspond; for instance, take $r=2.''5$ and $\alpha=16°$; $\tan\alpha=.29$. Then, from (b) and (c)

$$\frac{\tan\alpha}{k}=.0165,\quad k=17.6,\quad\text{and}\quad l=22.4.$$

From (d)

$$\frac{\tan\beta}{2m}(Ot)^2=\frac{\tan\beta}{m}\cdot\frac{(2.8)^2}{2}=\frac{\tan\beta}{m}\times 3.92=.125$$

and $\frac{\tan\beta}{m}=.0319$.

From (b) $\frac{m}{n\cos\beta}=1.43$.

Either n or m may be assumed, in accordance with the design of the front of the engine. Take $n=9''$, then

$$\frac{m}{\cos\beta}=12.87,\quad\text{and}\quad\frac{\tan\beta}{\cos\beta}=.41055=\frac{\sin\beta}{1-\sin^2\beta},$$

from which $\sin\beta=.439$ and $\beta=26°4'$.

This determines all the parts except the radius link.

In Equation 18, let $\cos\theta=\frac{r_2}{2l_2}$ and $\sin\theta=1$, then

$$-\frac{m}{n\cos\beta}\cdot\frac{kr}{l}\cdot\frac{r_2}{2l_2}=\frac{m}{n\cos\beta}\cdot\frac{l+k}{2l^2}r^2\tan a+\frac{m}{n\cos\beta}\cdot\frac{r^2}{2l_1}$$

$$-\frac{m}{n\cos\beta}\cdot\frac{(l+k)r^2}{2q_1l}+\frac{\tan\beta}{2m}\cdot\xi^2, \qquad (24)$$

since ξ is here equal to the lap.

Under the supposition that β was equal to zero, v_1v_2 formed the correction for $\frac{1}{2}$ cut-off. On the diagram $v_1v_2 = -.335$, which is exactly equal to the second part of Equation (24), since that forms the correction when $\beta = 0$.

Therefore,

$$\frac{m}{n}\cdot\frac{k+l}{2l^2}r^2\tan a+\frac{m}{n}\cdot\frac{r^2}{2l_1}-\frac{m}{n}\cdot\frac{(l+k)r^2}{2q_1l}=-.335,$$

from which $q_1 = 16''$

and $r_1=\frac{q_1}{\cos a}=16.''64.$

In the figure, lap circles are drawn to suit the modified travel, and the complete distribution of steam is shown.

The method shown in Fig. 11 may be employed after the parts have been laid down, as a check. It will be seen that considerable variation may be made in the various details, and their effects on the general distribution studied by these diagrams.

A slight angular advance may be given to the eccentric with advantage, sacrificing at the same time equality of lead for different cut-offs. It will be found to improve the distribution very much otherwise. The direction of this angular advance will depend on the position of the reversing quadrant and the connections of the eccentric lever, and must be determined for the special case. The whole gear may be designed, and the valve curves drawn without angular advance, and then a new center line of the engine drawn through O, making the required angle with OX. It must be drawn to increase the lead from full to mid-gear. This will determine its direction from the crank, and enable its effect to be studied on the diagrams.

The Marshall gear is the simplest of the radial valve gears. Its chief recommendation lies in the small number of working parts and the excellent distribution of steam, but the Stephenson

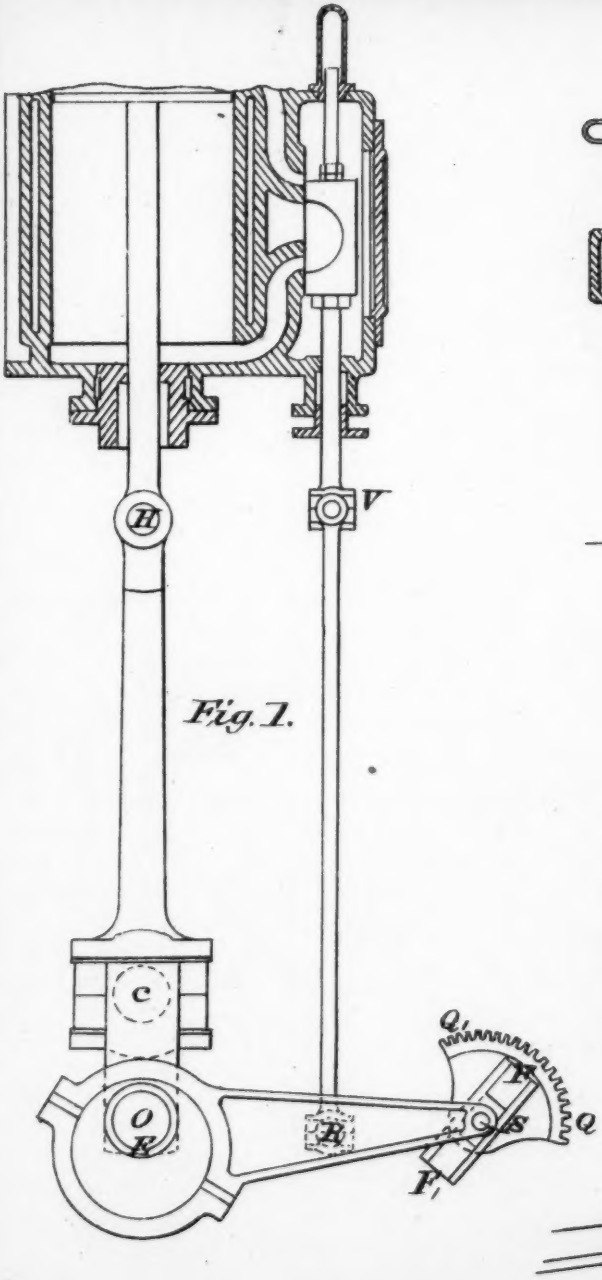


Fig. 1.

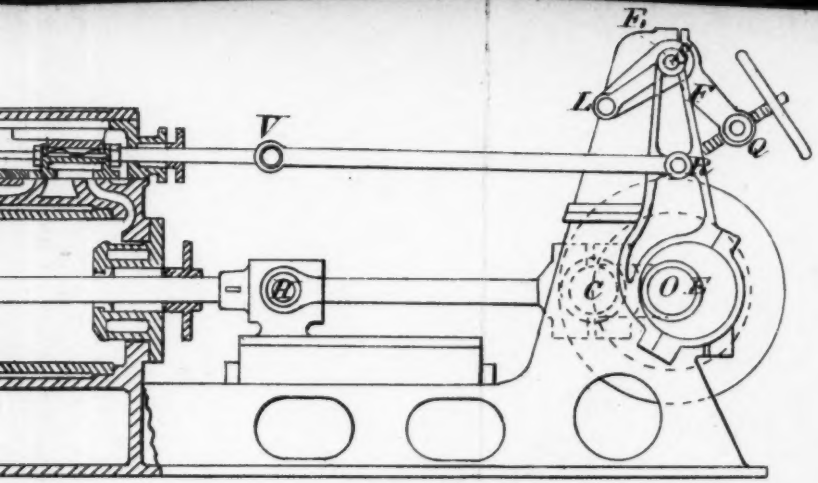


Fig. 2.

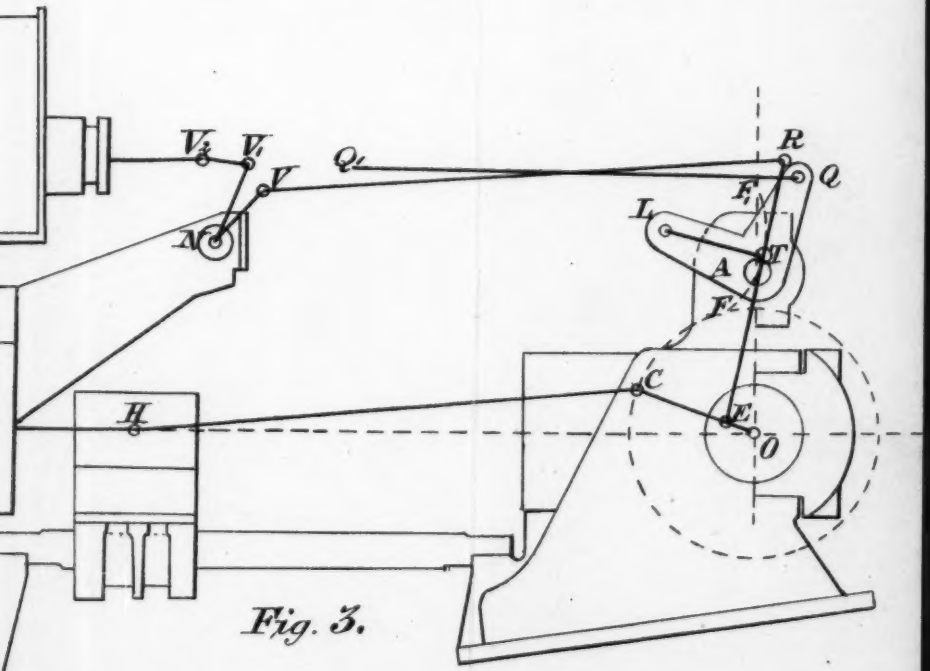


Fig. 3.

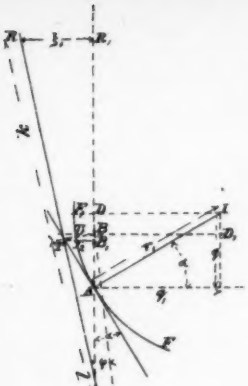


Fig. 4.

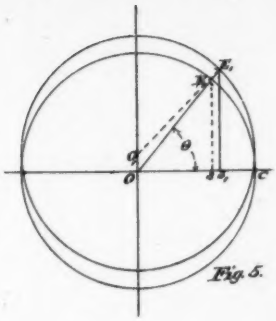


Fig. 5.

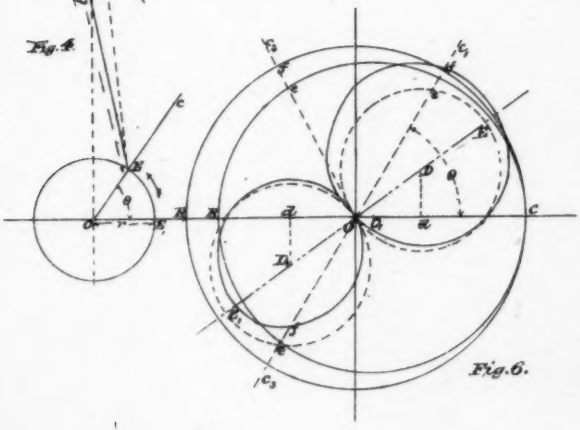


Fig. 6.

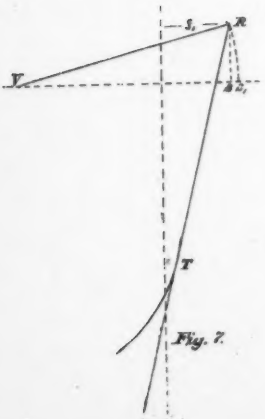


Fig. 7.

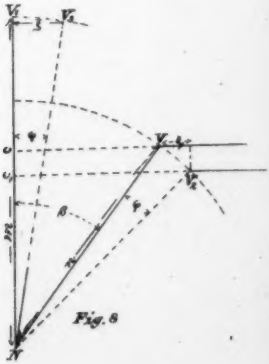
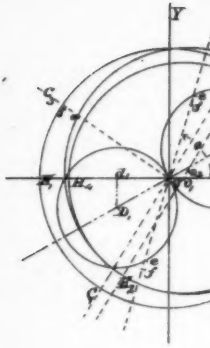
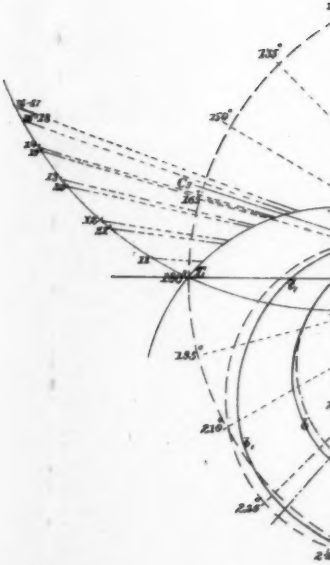


Fig. 8


Y
S
H
D
C


230°
250°
235°
210°
230°

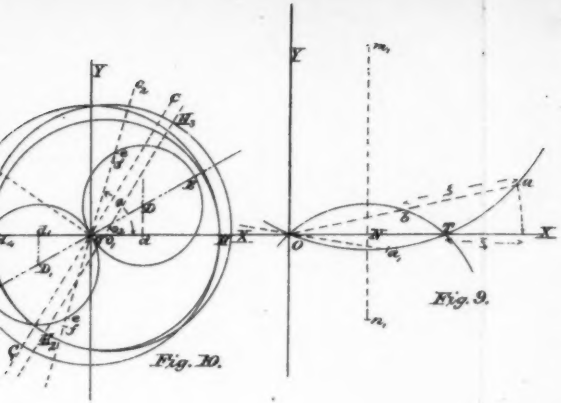


Fig. 10.

Fig. 9.

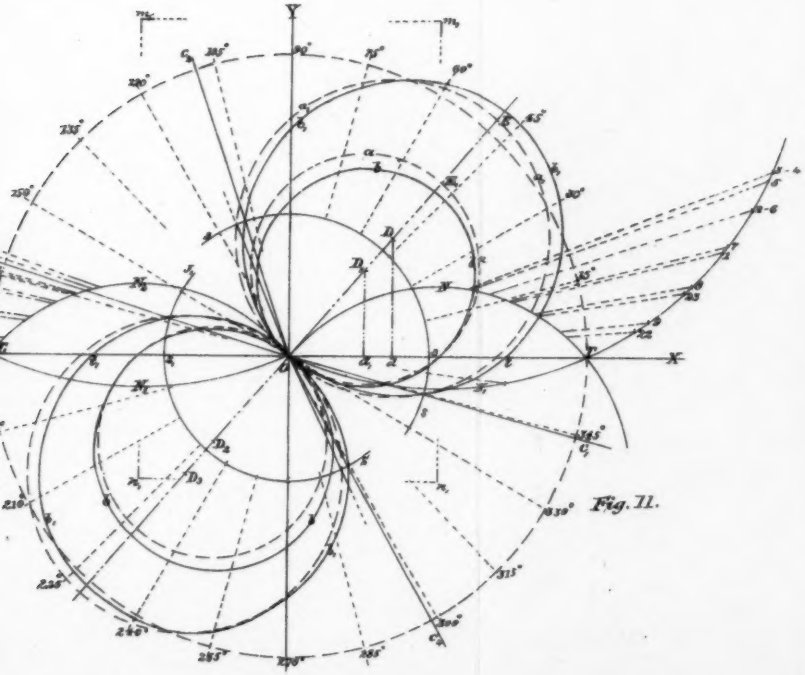


Fig. 11.

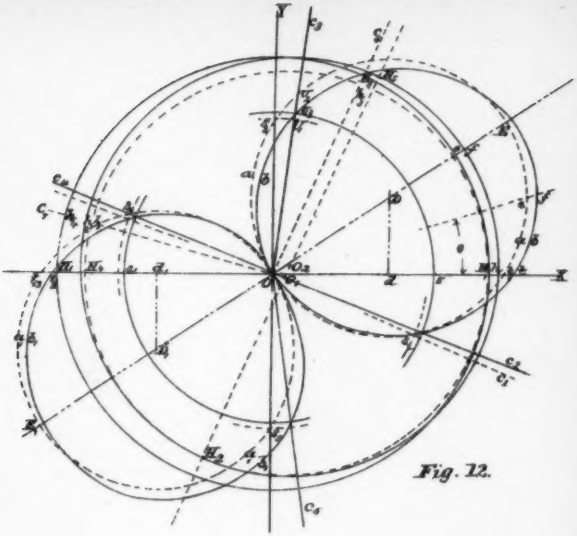


Fig. 12.

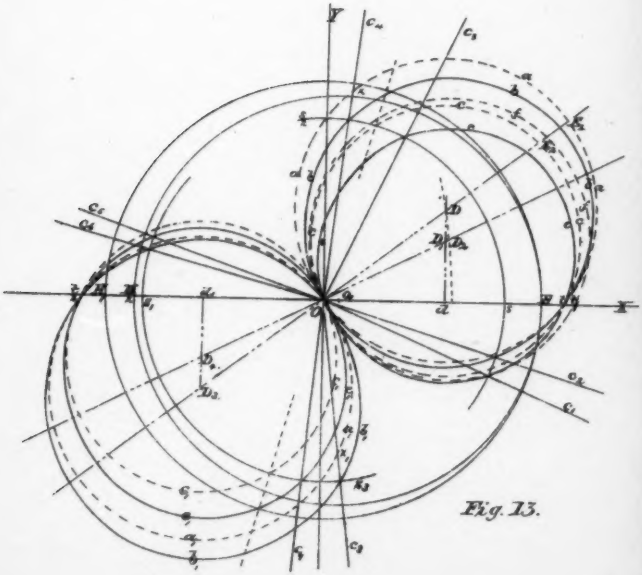


Fig. 13.

link may be designed to accomplish almost as much. In all the figures, the dotted circles show the valve motion due to the common eccentric, and afford a good comparison with the Marshall gear curves. See D_2, Fig. 13, for the center of the valve circle *f–f–f* of Stephenson link, linked up to correspond with the circle D_1. The difference in lead and general distribution is shown at once.

The lead may be made practically constant with the link by giving the backing eccentric greater angular advance than the go-ahead, but the distribution in backing will be distorted somewhat.

The Marshall gear is better, but there is also a sacrifice of the backing to the go-ahead motion. The constant lead is an advantage, but the equalization of cut-off is questionable. As may be seen from the valve curves, a greater opening must be provided on the outboard end in order to get equal quantities of steam into the cylinder on both ends.

For equal cut-offs, the actual time of opening will be different, owing to the angularity of the connecting rod, and the velocity of steam therefore different.

Unless the steam ports are very large, this becomes a serious matter with fast-running engines, as was shown on the trial of the *Charleston*. The lead, port openings and cut-offs were practically equal on the two ends, but the power was from 40 per cent. to 50 per cent. greater on the inboard end of the L. P. cylinder. This discrepancy was not so great in the H. P. cylinder where the velocity of steam was lower.

This difference of power may do very well with a vertical engine where the dead weight of the moving parts is taken by the steam, but is not so advantageous with a horizontal engine.

It would be well to apportion the cut-offs, or port openings, to obtain equal powers, or equal crank efforts on the two ends.

The compensating rockshaft adds something to the efficiency of the gear, but it also adds to the working parts.

The length over all of the engine may be decreased by placing the valves on the sides of the cylinders, to which the Marshall gear is particularly well adapted. The link does not lend itself readily to this arrangement without multiplication of parts.

II.

TEST OF THE WORTHINGTON PUMPING ENGINES BUILT FOR THE CITY OF NEW BEDFORD, MASSACHUSETTS.

By Chief Engineer Isherwood, U. S. Navy.

The experiment hereinafter described was made by the committee of experts, consisting of Messieurs Wm. Rotch, R. C. P. Coggeshall and Wm. R. Billings, appointed by "The New Bedford Water Board" to ascertain whether the pumping machinery furnished by the contractor, Mr. Henry R. Worthington, for the city of New Bedford, Massachusetts, fulfilled the guarantees of the contract, the acceptance of the machinery being conditional on the same.

By means of the information kindly supplied by Mr. Coggeshall, the very able superintendent of the New Bedford Water Works, and member of the Board of Experts, from his memoranda, in addition to what is contained in the report of that Board, the writer has been able to make from the data thus completed the analysis of the performance of the engines hereinafter given, the results of which he believes are worthy the attention of engineers.

The boilers which supplied the steam were the two vertical Corliss boilers described in Vol. I, No. 4, of the Journal, and which were afterwards specially tested by the same committee on the 16th and 17th of February, 1887, to determine their economic vaporization of water under the same conditions as existed during the pumping test of the 1st of June, 1886, and with the same kind of anthracite.

ENGINES.

The pumping was done by two horizontal, tandem, compound, condensing, steam jacketed engines, cutting off the steam in both the small and large cylinders, and were designed and built by

Henry R. Worthington, of the city of Brooklyn, New York, according to his well known compensating system. Each of the two engines consisted of a small and a large cylinder, so that during the experiment there were in use two small cylinders and two large cylinders, and as each engine operated one pump, there were two pumps in use. Each large cylinder was fitted with two piston rods, both on the same side of the piston. The small cylinder was placed between the pump and the large cylinder, the two cylinders being secured together by a strong casting of the form of a frustum of a cone. The piston rods of the large cylinder came through the head of that cylinder outside of the small cylinder, and were secured to a crosshead moving between the small cylinder and the pump, the axes of both rods being in the same horizontal plane.

Each small cylinder had one piston rod (on one side of the piston only), which passed on through the pump head and was fastened into the plunger or cylinder forming the piston of the pump. On this piston rod the crosshead was secured. The small cylinder was connected rigidly with the pump by means of wrought iron ties extending between the heads.

The valve gear was the same for both the small and the large cylinders, and is essentially a tappet arrangement. In this system two distinct engines are required, and they must be placed side by side, for, as there is no fly-wheel or rotating part on which an eccentric or crank could be placed to give the valves the proper movement, the valves of each engine have to be worked by the main movement of its companion engine, the pistons of one engine being at the end of their stroke while those of the other engine are at midstroke. The pumping is thus done direct, and there is a pause at each end of the stroke of the pistons. The movement of the pistons is nearly uniform, and at the end of each stroke it is arrested by the steam lead. This system of valve gear admits of such arrangement of the length of the stroke of the piston, that there need be absolutely no clearance in the cylinder, the pistons slightly striking the cylinder covers at the end of each stroke. Of course, close attention on the part of the engineer in charge is required for

producing this effect, as a slight change in the conditions of piston speed or pressures will either shorten the stroke or lengthen it so as to give too strong a blow to the covers of the cylinders. The point of cutting off the steam is not affected by the speed of the piston, but remains constant under all conditions unless changed by hand. During the experiment hereinafter described, there was no cylinder clearance, the pistons being kept slightly striking the cylinder covers throughout, consequently the waste spaces at the ends of the cylinder were only those in the steam passages.

Each cylinder had three valves: one an ordinary short slide, which admitted the steam and exhausted it in the usual manner as the pistons neared the end of their stroke. This slide valve worked horizontally on the upper side of the cylinder. The other two were vibrating plug valves placed one in the steam passage at each end of the cylinder, between the latter and the slide valve. The only office performed by these vibrating plugs was to cut off the steam at any point of the stroke of the piston to which they might be set. The slide valve was balanced by a piston on its top moving steam tight in a short cylinder cast with the valve, and suspended from a fixed journal above by a link connected with another journal on the piston. A hole drilled through the valve from the space beneath the piston into the exhaust cavity completed the design.

At each end of each cylinder there were two steam passages parallel to each other, in the outer one of which the vibrating plug valve was placed, the slide valve working over the ports of the four passages. At the commencement of the stroke of a piston, only the outer steam passage received the boiler steam, the slide valve and the vibrating plug valve being open. As the piston advanced it uncovered from beneath the second steam passage, the upper port of which was closed by the slide valve, so that both passages were filled with boiler steam when the vibrating plug or cut-off valve closed; consequently, the steam expended from the boiler when the cut-off valve closed was that due to the space displacement of the piston up to that point plus the entire content of the inner steam passage, and the content of

so much of the outer steam passage as was comprised between the cut-off valve and the cylinder. This was the bulk of steam that underwent expansion. When the piston reached the end of its stroke, the upper port of the outer steam passage was closed by the slide valve, and the upper port of the inner steam passage was beginning to open into the exhaust cavity of the slide valve; then the vibrating plug or cut-off valve opened, by which operation the entire content of the two steam passages was put in common with the interior of the cylinder, so that the bulk of steam exhausted at the end of the stroke of the piston was that of the space displacement of the piston per stroke plus that of the entire two steam passages.

Both cylinders were steam jacketed on sides and ends, the only part unjacketed being the portion of the sides covered by the valve chest. The water of condensation from the jackets drained into a closed receptacle beneath the cylinders whence it was pumped direct into the boilers by a small independent steam pump. This water was trapped to prevent the passage of steam.

The condenser was a jet condenser, and the air pump was distinct from the engine and worked by a small independent steam cylinder. The two feed pumps were also distinct and worked each by a small independent steam pump.

The exhaust steam from the air pump engine, from the jacket condensation engine, and from the engines of the two feed pumps, discharged into the hot well, together with the water discharged therein by the air pump, so that the temperature of this latter water was correspondingly increased by the heat in the exhaust steam described. From the water of the air pump discharge, thus additionally heated, the feed water was taken for the boilers, with the exception of the portion of the feed water taken from the steam jackets and due to the condensation of the steam therein, but the former portion of the feed water, before entering the boilers, was passed through a heater placed in the boiler flue, and surrounded by the gases of combustion after their escape from the boilers, and in this heater its temperature was raised still higher before entering the boilers. The temperature of the portion of the feed water derived from the steam

jackets was, of course, the temperature of the water in the boilers, only the latent heat being taken out of the steam undergoing condensation in the jacket.

A receiver, placed between the small and large cylinders, had the capacity of the former. All the external surfaces of the cylinders, receiver, steam pipes, etc., were well covered with non heat-radiating substances.

The water pump, one for each engine, consisted of a long piston or plunger moving in two chambers horizontally through a packed diaphragm at its center, separating them. These chambers, one at each end of the pump, were the receiving valve chambers. The delivering valves for both ends of the pump were in one chamber, immediately above the chambers of the receiving valves. The valves were circular, numerous, small, and controlled by springs. A large air vessel, common to both engines was placed above the delivering valve chambers.

The peculiarity of this engine is that, although it has no flywheel, and that it pumps by the direct action of its steam pistons upon its pump piston, the *net* steam pressure on the pistons being exactly equilibrated at every point of the stroke by the pressure of the water resistance, yet the steam can be used with any measure of expansion desired. This effect is accomplished by what may be termed a pneumatic equalizer or air compensator, constructed and applied as follows:

The principle of the equalizer is that of alternately compressing and expanding air in very small cylinders by the movement of the steam pistons of the small and large cylinders of the engine in conjunction with the simultaneous use of steam expansively; the compression of the air taking place during the portion of their stroke made by the steam pistons before expansion commences, and the expansion of the air taking place during the portion of their stroke through which the steam expands. The result due to a proper balancing of the steam and air pressures is such that if a diagram be made with the sum of the pressures of the steam and air at each point of the stroke of the pistons laid off by scale as ordinates perpendicular to a straight line base, the air pressures being reduced in proportion to the

capacity of the air cylinders to that of the steam cylinders, the free ends of the ordinates when joined would form a straight line nearly parallel to the base.

The apparatus embodying the above idea and attached to each engine consists of two very small compensating cylinders (7 inches diameter and 6 inches stroke of piston), oscillating on trunnions, one of which is hollow with its cavity in common with the interior of the cylinder. These cylinders are closed at one end and open to the atmosphere at the other end; that is to say, they are single acting cylinders, only one side of their pistons being used. The piston rods are connected to a crosshead which, in its turn, is secured to the piston rod of the pump extended through the back head or cover of the pump. This crosshead moves in guides on supporting brackets bolted to the pump cover. The included angle between the axes of the two compensating cylinders when the steam pistons are at either end of their stroke is about 69 degrees. Above the cylinders, and connected with them through their hollow trunnions, is a closed storage tank or reservoir. The compensating cylinders and their hollow trunnions are in common with this reservoir, and are filled, together with the lower portion of the reservoir, with water, which serves as a packing to the piston and hollow trunnions, and to receive the pressure of the alternately compressed and expanded air that occupies the upper portion of the reservoir above the water. The water is admitted from the main through a small pipe fitted with a cock or screw valve, and the air is kept at the required quantity and pressure by a small pump as needed. The compensating cylinders thus act as accumulators, and perform the functions of fly-wheels. Supposing the steam pistons at one end of their stroke and just commencing to move; then, as the crosshead of the compensating cylinders moves with them, the pistons of those cylinders compress the air in the reservoir, and at the same time the movement reverses the position of the axes of the cylinders by oscillating them on their trunnions. As soon as this reversal passes the perpendicular, the pistons reverse their motion and the compressed air begins to expand, continuing its expansion until the end of the stroke of the steam pistons.

When the latter reverse their movement for their succeeding stroke, the same operations are repeated by the compensating cylinders, and so on.

The compensating cylinders are, of course, placed one above and one below the axis of the steam cylinders, the axes of the compensating cylinders making equal angles with the axis of the steam cylinders, so that there is no side or cross strain on the crosshead or its guides.

The jet condenser and air pump are separate from the engines and serve for both. The air pump is horizontal, double-acting, and has a plunger or piston 8½ inches in diameter worked direct by a steam cylinder of 6 inches diameter and 10 inches stroke of piston.

The diameter of the receiving pipe of the pumps was 24 inches, which was also the diameter of the delivering pipe, the latter discharging into a main of 30 inches diameter and 1,879 feet length conveying the water to the distributing reservoir.

The following are the dimensions and proportions of the parts of one engine required to be known. Two such engines were in use during the experiment. The pistons had no clearance, but touched the cylinders ends.

Number of small cylinders..... 1
Diameter of the small cylinder..... 18 inches.
Diameter of the piston rod of the small cylinder..... 3¼ inches.
Net area of the piston of the small cylinder..... 250.322 square inches.
Stroke of the piston of the small cylinder..... 26 inches.
Space displacement of the piston of the small cylinder, per stroke..... 3.766 cubic feet.
Steam space in the steam passages at one end of the small cylinder filled with steam when the cut-off valve closed..... 0.128 cubic foot.
Steam space in the steam passages at one end of the small cylinder filled with steam when the exhaust port opened..... 0.188 cubic foot.
Per centum which the steam space in the steam passages at one end of the small cylinder filled with steam when the cut-off valve closed, is of the space displacement of the piston per stroke..... 3.4
Per centum which the steam space in the steam passages at one end of the small cylinder filled with steam when the exhaust port opened, is of the space displacement of the piston per stroke..... 5.0
Area of the steam port of the small cylinder (2 by 4 inches =)..... 8 square inches.
Number of large cylinders..... 2.
Diameter of the large cylinder..... 36 inches.

Diameter of the piston rods (2) of the large cylinder.....2½ inches.
Net area of the piston of the large cylinder.....1012.969 square inches.
Stroke of the piston of the large cylinder.....26 inches.
Space displacement of the piston of the large cylinder per stroke..... 15.241 cubic feet.
Steam space in the steam passages at one end of the large cylinder filled with steam when the cut-off valve closed..... 0.610 cubic foot.
Steam space in the steam passages at one end of the large cylinder filled with steam when the exhaust port opened..... 1.006 cubic feet.
Per centum which the steam space in the steam passages at one end of the large cylinder filled with steam when the cut-off valve closed, is of the space displacement of the piston per stroke..... 4.0
Per centum which the steam space in the steam passages at one end of the large cylinder filled with steam when the exhaust port opened, is of the space displacement of the piston per stroke..... 6.6
Area of the steam port of the large cylinder (2 by 10 inches).....20 square inches.
Interior length of both cylinders, between heads..... 32½ inches.
Face of the pistons of both cylinders..... 6½ inches.
Length of steam pipe from boilers to engines.....194 feet.
Ratio which the net area of the piston of the large cylinder is to the net area of the piston of the small cylinder, the latter being taken as unity..... 4.0466
Ratio which the space displacement of the piston of the large cylinder per stroke plus the steam space in the steam passages at one end of the large cylinder filled with steam when the exhaust port opened, is to the space displacement of the piston of the small cylinder per stroke plus the steam space in the steam passages at one end of the small cylinder filled with steam when the exhaust port opened, the latter being taken as unity.... 4.1083
Number of water pumps (double-acting).....1.
Diameter of the water pump..... 24 inches.
Diameter of the piston rod on one side of the water pump piston..... 3¼ inches.
Diameter of the piston rod on the other side of the water pump piston... 2½ inches.
Net area of the piston of the water pump.....445.788 square inches.
Stroke of the piston of the water pump.....26 inches.
Space displacement of the piston of the water pump per stroke.....6.707 cubic feet.
Ratio which the net area of the piston of the large cylinder is to the net area of the piston of the water pump, the latter being taken as unity.....2.272

THE EXPERIMENT.

The experiment was conducted in the following manner, namely:

The fires were started in the furnaces of the two Corliss boilers used at 7.45 A. M., June 1, 1886, and the engines were started at 8.50 A. M. and kept steadily in operation until noon, when the condition of the fires was carefully examined, the height of the water in the glass gauge of the boiler accurately noted, and the

experiment held to commence. The counter was also noted exactly at noon. The experiment continued 12 hours and 5 minutes consecutively, and was terminated with the water at the same height in the boiler and the fires in the same condition in the furnaces as at the commencement. The counter was noted at precisely the end of the experiment. As all the conditions remained nearly exactly the same during the experiment, the observed data varied only to the most trifling extent. The point of cutting off the steam in the cylinders, of course, remained unaltered.

The fuel consumed was hand-picked anthracite, and the two furnaces were fired alternately at 30 minutes intervals. The rate of combustion was so slow that the furnaces scarcely required cleaning during the short time of the experiment.

During the experiment the number on the counter and the pressure of the steam in the boiler and at the engine were noted every ten minutes.

Half-hourly observations were noted of the height of the barometer, of the vacuum gauge on the condenser, of the water pressure gauge on the main, of the height of water in the boilers, of the height from the center of the water gauge to the level of the water in the well of the pumps. There were also noted at half-hourly intervals the temperature of the water in the well of the pumps, of the external atmosphere, of the air in the engine room, of the water discharged by the air pump into the hot well of the engines, of the portion of the feed water drawn from the hot well when entering the heater and when leaving it, of the superheated steam in the boilers and of the steam at the engines, and of the gases of combustion in the flue before entering the heater and after leaving it.

The water pressure gauge on the main was tested by comparison with a standard gauge of the Crosby Steam Gauge Company, and was found to be accurate within the limits registered during the experiment.

The height of water in the well of the pumps was obtained by two large float gauges tested previous to the experiment, and also by measurement with a graduated stick.

The temperature of the steam was obtained as follows: Two iron cups filled with mercury were screwed into the steam pipe one near the boiler, and the other near the engine. These cups were deep enough to extend nearly across the pipe. Thermometers were placed in the mercury with their bulbs and as much of their stems as possible covered by it.

The barometer was an aneroid, placed in the engine room. It had been tested in the Royal Observatory at Greenwich.

Indicator diagrams were frequently taken from the steam cylinders and from the water pumps.

The "slip" of the pumps, or their loss of action, was measured at a weir 4 feet in length, for considerable intervals at a time, and did not exceed 2 per centum at the utmost.

The anthracite consumed was weighed with the greatest possible accuracy, and its per centum of dry refuse in ash, clinker, and unburned fragments falling through the grates was determined as nearly as possible, the experiment being a portion of the regular pumping work of the engines. The rate of combustion was as high as the boilers would allow. If this rate was exceeded foaming or priming commenced.

One portion of the feed water was the water of condensation supplied by the steam jackets and pumped directly into the boilers. This quantity was not measured during the experiment, but its weight was known from measurements taken at other times, under the experimental conditions, to be 286 pounds per hour, which has, therefore, been assumed to be the weight of feed water supplied during the experiments to the boilers by the condensations in the steam jackets. The temperature of this portion of the feed water was, of course, the temperature of the water in the boiler, only the latent heat having been taken out of the jacket steam by the condensations.

The remaining portion of the feed water was drawn from the hot well, and was measured by a Worthington meter, through which it passed on its way to the boiler. The correction for the meter was carefully ascertained and applied.

The temperature of the condenser water as delivered by the air pump into the hot well was 83.60 degrees Fahrenheit. The

water had its temperature in the hot well raised to 179.04 degrees Fahrenheit by the exhaust steam of the auxiliary engines discharging therein. After leaving the hot well and before entering the heater, the portion of the feed water was measured by the meter. After passing the meter it entered the heater with the temperature of 179.04 degrees, and leaving the heater entered the boiler with the temperature of 231.08 degrees. The difference between the last two temperatures was imparted by the heater. The mean temperature of the entire feed water entering the boiler was 242.54 degrees Fahrenheit.

Of the entire number (34,507,924.7335) of Fahrenheit units of heat imparted to the entire feed water in the boiler, 31,510,305.18 were imparted to the water measured by the meter, and 2,997,619.5535 were imparted to the water furnished by the steam jackets.

In the following table will be found the data and results of the experiment, the quantities being grouped and numbered for facility of reference.

EXPLANATION OF THE TABLE.

The experimental quantities are so fully described on the lines of the following table, that but little explanation will be needed, and only as regards a few items, but some space may be devoted to the consideration of their causes and results in particular cases.

Total quantities.—The total number of double strokes made by the steam pistons of the engines, given on line 2, is the same as the number of revolutions, a "go and come" of a piston corresponding to a "revolution."

The total number of pounds of feed water pumped into the boilers from the hot well, given on line 3, is the portion of the feed water measured by the meter, and entering the boiler with the temperature on line 42.

The total number of pounds of feed water additional to the number of pounds given on line 3, which was supplied to the boilers by the water of liquefaction or condensation in the steam jackets, and which entered the boilers with the temperature of the boiler water, is given on line 4.

The total number of pounds of feed water entering the boilers from both of the above sources, is given on line 5, being the sum of the quantities on lines 3 and 4.

Line 6 contains the total number of Fahrenheit units of heat imparted to the feed water in the boilers, above the temperature of that water when entering the boilers.

Line 7 contains the per centum which the quantity on line 4 is of the quantity on line 5. The quantity on line 7 is the per centum which the water of condensation drawn from the steam jackets is of the entire feed water vaporized in the boiler.

Line 8 contains the number of pounds of feed water vaporized in the boilers to supply the cylinders alone with steam; that is to say, exclusive of the number of pounds of feed water vaporized in the boilers to supply the cylinder jackets with steam. The quantity on line 8 is the same as that on line 4, and is the weight of steam that entered the small cylinders during the experiment.

On line 9 is the total number of pounds of anthracite consumed in the furnaces, and on line 10 is the total number of pounds of refuse from the anthracite in ash, clinker and small pieces of unconsumed anthracite falling through the grates. The difference of these two quantities is given on line 11, and called "combustible," in which term is included the hygrometric moisture of the crude anthracite. On line 12 will be found the per centum which the quantity on line 11 is of the quantity on line 10.

Line 13 contains the total number of cubic feet of water delivered by the pumps into the main, calculated from the capacity of the pumps and the number of strokes made by their pistons, less 2 per centum allowed for incomplete filling and regurgitation past their valves and pistons.

On line 14 is the number of United States gallons of water delivered by the pumps during the experiment. This quantity was obtained by dividing the number of cubic inches in the quantity on line 13 by 231, the number of cubic inches in a United States gallon.

Line 15 gives the total number of foot pounds of work done by the engines external to them, calculated as follows: The water pressure in the main, at the point where the pressure gauge,

whose indication in pounds per square inch above the atmosphere is given on line 28, was attached, plus the water pressure in pounds per square inch due to the vertical height of this gauge above the level of the water in the well of the pumps, as given on line 33, is the pressure external to the pumps against which they discharged; hence the external work in foot pounds done by the engines will be given by multiplying the quantity on line 13 by 144 (the number of square inches in a square foot), and the product by the sum of the quantities on lines 28 and 33. The final product is the quantity on line 15, or the "duty" of the engines, corresponding to the work of an engine in foot pounds done at the brake. This work includes not only the lifting of the water from its level in the well of the pumps to its level in the reservoir, but also the resistance overcome of the wetted interior surface of the main, and the resistance overcome due to the sinuosities of the main.

Engines.—The steam pipe of 194 feet length, in conjunction with a partly closed throttle, made about 5¼ pounds per square inch difference between the pressure in that pipe near the boiler and near the cylinder (lines 17 and 18). The throttling (line 19) was necessary to prevent the boilers from foaming or priming.

Lines 24 and 25 give, respectively, the fractions of the stroke of the pistons of the small and large cylinders completed at the instant the cut-off valves closed. These fractions are the means obtained from all the indicator diagrams taken; the point of cutting off was taken to be the point at which the convex curve, due to the gradual closing of the steam port, reverses into the concave curve of the expansion pressures. There was no cushioning of the steam in the cylinders, and no release until the pistons reached the end of their stroke.

Line 26 shows the number of times the steam was expanded in the small cylinders, obtained by dividing the content of the cylinder plus the content of the waste space at one end of it, by the content of the cylinder at the point of cutting off plus the content of the waste space at the point of cutting off.

Line 27 shows the number of times the steam was expanded in the engine, obtained by multiplying the quantity on line 26

by the ratio which the content of the large cylinder plus the content of the waste space at one end of it is of the content of the small cylinder plus the content of the waste space at one end of it.

Pumps.—Line 28 shows the pressure of the water in the main in pounds per square inch above the atmosphere, as given by a gauge at the level of the axes of the pumps, and line 29 shows the pressure in pounds per square inch above the atmosphere on the pistons of the pumps during their delivery stroke, as given by an indicator. The difference between these two pressures, given on line 34, is the pressure in pounds per square inch required to lift the delivering valves of the pumps and to force the water through the openings in the valve seats. This pressure, even when the valve-seat openings are abundantly large, is, of course, much greater than what is due to the weight of the valves, because there is not only this weight to be lifted, but the vertical direction of the water current through the valve seats has to be changed to the horizontal direction, in order that the water pass under the lifted valves, and then, when free from their constraint, the water direction has to be again changed to the vertical, and these changes have to be made against the opposing pressure in the main.

The proper arrangement for pump valves, were it practicable, would be to make them slide valves, worked by a cam movement, so that as the pump piston commenced its stroke, they could be rapidly drawn sideways from the valve seat openings, and the water allowed to pass through without change of vertical direction. Of course, slide-valves would have a friction from which the vertically lifting valves are free, so that the net gain would be the difference between the pressure required with the lifting valves and the pressure required to overcome the friction of the slide valves. The lifting valves, however, have some adhesion to their seats to be overcome.

During the receiving stroke of the pump piston, the pressure on it was 8.28 pounds per square inch below the atmosphere, as given by the indicator, and this is the quantity shown on line 30. Now the pressure due to the vertical height of 17.56 feet

of water column between the level of the water in the pump well and the pressure gauge on the main, was, for the experimental temperature of the water, 7.60 pounds per square inch, as shown on line 33; consequently, the difference between the 8.28 and the 7.60, which is 0.68, is the fraction of a pound per square inch required to lift the receiving valves of the pump and to force the water through the openings in their seats. From the foregoing will be seen that the pressure per square inch of the pistons of the pumps required to overcome the resistances due to the receiving and delivering valves, was $(2.50 + 0.68 =) 3.18$ pounds. As the ratio which the net area of the pistons of the large cylinders of the engine is to the net area of the pistons of the water pumps, is 2.272 to unity, the stroke of all the pistons being the same, the above pressure of 3.18 pounds per square inch on the pump piston is equivalent to 1.168 pounds pressure per square inch of the large cylinder piston. Now, the net pressure on the pistons of the large cylinders alone, corresponding to the net horses-power developed by the engines, being 25.745 pounds per square inch, line 81, there were absorbed by the pump valves resistances $(\frac{1.168 \times 100}{25.745} =) 4.5368$ per centum of the net horses-power developed by the engines, a very serious quantity, which should be lessened if practicable.

Line 31 contains the indicator pressure in pounds per square inch on the pistons of the pumps, as given by an indicator directly applied. The lower edge of the indicator diagram, representing the pressure of the incoming water, was quite serrated or ragged, but the upper edge was very smooth and regular, with a well-marked concave curvature towards the straight atmospheric line, showing that the pressure on the pump piston gradually increased from the beginning of the stroke to the center thereof, and as regularly decreased from the center of the stroke to the end thereof, the pressure at the center being a couple of pounds per square inch greater than at the ends of the stroke. Of course this slight difference in the pressures on the pistons of the pumps was due to corresponding variations in the speed of those pistons, and would have been more marked but for the action of the air in the large air vessel above the delivering valves chamber.

The height in feet between the levels of the water in the pump well and in the reservoir was not ascertained during the experiment; had it been, the difference between the pressure due to this statical head and the experimental or dynamical head, as shown by the gauge on the main, would have given by their difference the pressure required to overcome the resistance of the interior wetted surface of the main and of the changes of the water direction in the same. The pressure on line 31 is that which overcomes the dynamical head and the resistance of the pump valves, and it acts during the entire stroke of the pistons of the pump.

Line 36 shows the number of horses-power developed by the pistons of the pumps, calculated for the pressure on line 31, and for the speed of these pistons per minute.

Temperatures.—Line 40 gives the temperature of the water discharged by the air pump from the condenser into the hot well of the engines, and line 48 gives the temperature of the same water after it had received the heat of the exhaust steam of the small auxiliary engines. The rise of temperature due to this cause was 95.44 Fahrenheit degrees, a most important quantity, equivalent to 7.794 per centum of the heat above the temperature of the water discharged by the air pump, imparted to the total feed water vaporized in the boilers, although the rise of temperature was only in the portion of the feed water measured by the meter, and exclusive of the portion supplied from the steam jackets. Had the air pump and feed pumps been worked by the main engines, there would have been no such accession of heat to the feed water, and it probably greatly more than equalized the greater heat cost of the power required to work the auxiliary engines than if their work had been done by the main engines. Consequently, there would have been no heat saved by the main engines doing the work of the auxiliary engines. If the condenser, instead of being a jet condenser, had been a surface condenser, in which case the water delivered by the air pump into the hot well would have been comparatively little, being only the water of condensation of the exhaust steam of the main engines, as it would not have included the injection water, and if

the water delivered from the surface condenser had been of such weight and temperature as would have enabled it to condense the exhaust steam from the auxiliary engines, then the heat cost of the work of those engines would have been only what was theoretically due to the development of their power, for there would not have been any rejected heat. In other words, the entire heat in the steam entering the cylinders of the auxiliary engines would have been utilized, either in producing the power developed in them or in heating the feed water.

When the portion of the feed water that was measured by the meter entered the heater, it had the temperature of 179.04 Fahrenheit degrees (line 41), and when it left the heater it had the temperature of 231.08 degrees, the difference being 52.04 degrees, equivalent to 4.51 per centum of the heat above the temperature of the water discharged by the air pump, imparted to the total feed water vaporized in the boilers. Thus, by means of the heater and the exhaust steam of the auxiliary engines discharged into the hot well, there were saved (4.51 + 7.79 =) 12.30 per centum of the heat that would have had to be imparted to the feed water had the portion of it measured by the meter entered the boiler with the temperature of 83.6 degrees with which the mixed water of condensation due to the exhaust steam from the main engines and the injection water were discharged by the air pump from the condenser into the hot well of the engines. Had the water in the hot well not been heated by the exhaust steam of the auxiliary engines, the heater would have been more efficient, because the temperature of the water entering it would have been a great deal lower, and because it acts only by the difference between the temperature of the water entering it and the temperature of the gases of combustion enveloping it.

For maximum economic effect from a given area of water heating surface, with equal steam pressure, equal temperature of feed water and equal rate of combustion of fuel, all boilers should be furnished with heaters containing about 40 per centum of the aggregate water heating surface in boiler and heater. The surface should be arranged as long as practicable, and the gases of combustion should be made to pass successively over the sur-

face in the reverse direction to the flow of the feed water. The extreme limit to which the feed water in the heater can be advantageously heated is the temperature of the water in the boiler; practically, there should be a difference of not less than 50 degrees between the temperature of the water in the boiler and the temperature of the water entering the boiler from the heater.

The superior efficiency of the combined boiler and heater over a boiler alone, other things being equal, consists in the possibility in the first case of reducing the temperature of the gases of combustion to a lower degree than can be done in the second case, because the reduction of the temperature of those gases is wholly due to the temperature of the water on which they last act. With the boiler alone, that temperature is the temperature of the water or steam in it. With the boiler and heater combined, that temperature is the temperature of the feed water in the hot well.

With a boiler and heater whose heating surfaces are relatively as above described, the aggregate being twenty-five square feet of heating surface to one square foot of grate surface, the calorimeter being one-eighth of the grate surface, the rate of combustion being about 12½ pounds of coal per hour per square foot of grate, the temperature of the water in the boiler being about 340 degrees Fahrenheit, and the temperature of the feed water being about 100 degrees, the temperature of the gases of combustion will be reduced about 200 degrees lower by the boiler and heater combination than by the boiler alone, the same aggregate heating surface being employed in both cases, giving an increased economy of one-ninth of the fuel, a very serious quantity and worth much effort to obtain.

Of course, the economic value of the same heater will vary greatly with the conditions under which it is employed. If, for example, the boiler heating surface, in conjunction with the rate of combustion, was such that the temperature of the gases leaving it was but little above the temperature of the water in it, the heater would have but little margin to work on, and would give a proportionally small result. And if the steam pressure in the boiler was very low with a correspondingly low temperature of

the water in the boiler, and if the temperature of the feed water was high, the heater would give a still smaller and smaller result than before. But, on the contrary, the higher the boiler steam pressure and water temperature, the greater the rate of combustion, the less the heating surface in the boiler, and the lower the temperature of the feed water, the greater will be the economic result due to the employment of a heater. Like all physical problems, the outcome depends on the limitations, and varies largely with the conditions.

The water-heating surface in the experimental boilers, measured on the inside of the tubes, was 859.9322 square feet. The water-heating surface in the heater similarly measured was 183.2600 square feet. Number of square feet of water-heating surface in boiler per square foot of water-heating surface in heater, 4.6924. Number of square feet of grate surface, 37.1716. With these proportions and the low rate of combustion, the performance of the heater must be considered excellent.

The steam in the boilers was superheated 22.67 degrees Fahrenheit, namely, from 348.29 degrees, the temperature due to the boiler pressure had the steam been saturated, up to 370.96 degrees. But in passing through the well clad 194 feet long steam pipe, this superheating was nearly all lost, the temperature of the steam at the cylinders being 346.28 degrees, while the temperature of saturated steam of the pressure there is 345.77 degrees.

This loss of temperature, as well as the loss of pressure, experienced by *superheated* steam in passing through the steam pipe from the boiler to the cylinder, is universally ascribed wholly to radiation, whereas, in fact, but a very small portion of the temperature and none of the pressure are thus lost in a well protected pipe like the one in the experiment. In the case of *saturated* steam there is no loss of either temperature or pressure by the steam in passing through a steam pipe, however long and wholly unprotected by covering it may be. There is, indeed, loss of *heat* due to the radiation, which is evidenced in the liquefaction of a corresponding portion of the saturated steam in the pipe. But the supply of steam being equal to the de-

mand, this loss by radiation is made good in the boiler by the consumption of correspondingly more fuel, so that the pressure and the temperature of the non-liquefied steam in the pipe would be maintained. Nevertheless, the irrefutable evidence of the steam gauges does show that there is a loss of temperature and pressure in the steam pipe, and that these losses become more and more as the length of the pipe, as the velocity of the steam through it, and as the density of the steam in it, become greater and greater. But the same evidence will also show that for the same conditions these losses are exactly the same whether the pipe be thickly clad with non heat-radiating coverings or exposed nude. The experiment is easily made in any given case, and the result will be found as stated. Now, the admitted losses not being due to radiation, to what are they due? Why, simply to the expansion of the steam in the pipe necessary to the transportation of the same through it; for only by virtue of this expansion can the steam be conveyed by the boiler to the cylinder. The steam, being an elastic substance, cannot be pushed as a solid body would be, unaltered, the length of the pipe by the steam generated in the boiler, thus preserving pressure and temperature. It has to transport itself by its own expansion, accompanied, of course, not only by decrease of pressure, because, in order to expand, there must be less pressure at the delivering than at the receiving end of the pipe, and by decrease of temperature, because the work of the transportation must be done at the expense of the contained heat of the expanding steam. All that the boiler does in this case is to furnish the steam to the receiving end of the pipe, and this steam then propels itself through the pipe by its own elasticity, and in so doing there is a transmutation of its contained heat into the work of the propulsion in the ratio of one Fahrenheit unit of heat for every 789¼ foot pounds of work done. In the case of saturated steam the transmutation of its heat liquefies a portion of the steam corresponding to the work accomplished, but in the case of superheated steam—if the superheating is sufficiently great—there is no liquefaction, the work done being wholly at the expense of the superheat, as shown by the reduction of the tem-

perature of the expanding steam without liquefaction of any portion of it. If the superheating be not sufficient then liquefaction supplies the rest of the heat. In the experimental case, the superheating seems to have been just enough to furnish the steam in the *dry* saturated state to the cylinders. Of course, the superheating, though ceasing at the cylinders, was nevertheless economically valuable, as without it the steam furnished to the cylinders would have been *wet* saturated steam, with which the power in the cylinders would have been developed at greater cost of steam weight or fuel.

Owing to the leakage of air into the uptakes and flues of the boilers, the temperature of the gases of combustion in the chimney (line 48) was doubtless much lower than it should have been, but the small surface of the heater, with the feed water entering it at the high temperature of 179.04 Fahrenheit degrees, the rate of combustion being quite low, reduced the temperature of these gases 101.6 degrees (line 50), equivalent to an absorption of about 5⅔ per centum of the heat of combustion of the fuel.

Heat per Hour, and Duty.—Line 51 gives the number of pounds of feed water pumped into the boiler per hour, whose steam supplied the cylinders and their steam jackets; and line 53 gives the number of pounds of feed water pumped into the boilers per hour, whose steam supplied the cylinders alone. The difference of these quantities gives the number of pounds of feed water pumped into the boiler per hour, whose steam supplied the jackets alone. The quantities on lines 51 and 53 are the quotients respectively of the division of the quantities on lines 5 and 3 by the number of hours on line 1.

Line 52 contains the number of Fahrenheit units of heat above the temperature of the feed water in the quantity on line 51. And line 54 contains the number of Fahrenheit units of heat above the temperature of the feed water in the quantity on line 53.

Line 55 shows the "duty" of the engines. This quantity is the quotient of the division of the quantity on line 15 by the quantity on line 9, multiplied by 100.

The above "duty" represents the commercial efficiency of the system as a whole under the experimental conditions, and it will vary with these conditions. It includes the economic efficiency of the boiler and of the fuel, the skill and care of the fireman and engineer, the lubrication and keying of the journals, the degree of tightness of the packings, the vacuum in the condenser, the reciprocating speed of the piston, the savings of heat by heaters, and the utilization of the exhaust steam from auxiliary engines, the gains by steam superheating and steam jacketing, etc., as well as the efficiency of the type of engine, the excellence of its valve gear, the pressures between which its pistons work, and the measure of expansion with which the steam is used. The "duty" will vary with all and any of these with the same engine, so that the excellence of the type of engine alone cannot be appreciated from it.

In the case of purchasing pumping engines with guarantee of "duty," the contractor should furnish the entire plant, boilers and concomitants as well as engines, using a specified fuel and pumping a specified quantity of water during 24 consecutive hours per 100 pounds of crude fuel consumed without use of, or deduction for, the unconsumed portions falling through the grate, thus putting the bargain on a purely commercial basis. The water to be computed by the delivery of the pumps per stroke of their pistons, and the height to which it is lifted being the vertical distance from the level of the water in the pump well to the level of the water in the reservoir. The "duty" test and the "capacity" test to be one and the same. The "duty" is purely a commercial matter and has no engineering value.

Rate of Combustion.—Lines 56 to 61, both inclusive, give the rate at which the crude anthracite, and the portion remaining of it after deducting ash, clinker and small unconsumed pieces falling through the grate, were consumed per hour, and per square foot of grate and heating surface per hour.

Steam Pressures in the Small Cylinders per Indicator.—Lines 62 to 70, both inclusive, give the various steam pressures in the small cylinders per indicator. They are the means from all the

indicator diagrams taken. The steam was not released until the end of the stroke of the pistons, and it was not cushioned. The indicated and the net pressures (lines 67 and 68) are given in pounds per square inch of the pistons, the indicated pressure being above the back pressure (line 65), and the net pressure being the indicated pressure less 4.55 pounds per square inch taken as the pressure on the pistons of the small cylinders required to work the engines *per se* or unloaded. All the other pressures are in pounds per square inch above the zero of pressure. The total pressure (line 69) is the sum of the pressures on lines 67 and 65. The pressure on line 70 is the mean pressure of the expanding steam from the closing of the cut-off valves to the end of the stroke of the pistons. The steam pressure on line 63 was taken at the point where the convex curve of the indicator diagram just before the closing of the cut-off valve changes into the concave curve of the expanding steam from the instant that valve closes.

Steam Pressures in the Large Cylinders per Indicator. Lines 71 to 79, both inclusive, give the various steam pressures in the large cylinders at corresponding points to those just described for the small cylinders. The net pressure (line 77) is 1.132 pounds per square inch less than the indicated pressure (line 76), that being the pressure required to work the engines *per se* or unloaded.

The horses-power required to work the engines *per se* or unloaded, is given experimentally by the difference between the aggregate indicated horses-power developed by the engines (line 95), and the aggregate horses-power developed at the pistons of the pumps (line 36). This difference of power is divided between the small and the large cylinders in the ratio of the net areas of their pistons, and the pressures corresponding to these apportionments of power are the differences in the respective cases between the indicated and the net pressures on the pistons of the small and the large cylinders.

As the pumping was done directly—that is, by the direct action of the steam pistons upon the pistons of the pumps—the straight line movement of the steam pistons not having to be

converted into the rotary movement of a crank, and then reconverted into the straight line movement of the pistons of the pumps, there was saved the friction of the load, which is always lost when power is passed from a straight line to a circular direction, because of the friction due to the constraining action of the mechanism. Hence, in all pumping engines, and, elsewhere, too, when possible, the work should be done without change of direction; the forcing of the power from the straight line to the circular direction, causing a loss of about 7½ per centum of the net power developed by the pistons of an engine. Pumping by means of a vibrating beam has the same objection for the same reason and to the same extent.

The total pressure on the pistons (line 78) is the sum of the indicated (line 76) and of the back pressure (line 74). This quantity is the same as that on line 79, because all the steam in the large cylinders is expanded steam, that fact not being changed by the use of a cut-off valve on these cylinders, and because the steam was not cushioned.

Equivalent Pressures on the Pistons of the Large Cylinders.— To constitute a compound engine two cylinders of different capacities are required, with the steam interacting upon their pistons, consequently there are two sets of mean cylinder pressures, and as these sets have no fixed relation, there results the necessity for the purposes of comparison to reduce the two sets to one. This is easily accomplished for the indicated and net pressures by dividing these pressures in the small cylinder by the ratio of the net areas of the pistons of the small and large cylinders, and adding the quotients respectively to the experimental indicated and net pressures in the large cylinder. The stroke and speed of the pistons of both cylinders is here supposed to be the same. There is thus obtained what would have been the indicated and net pressures had the indicated and net horses-powers developed by the engine been developed in the large cylinder alone. In this manner the quantities on lines 80 and 81 were computed. In the present case, in which there was no cushioning, the total equivalent pressure above zero on the piston of the large cylinder can be computed by dividing the number of foot-pounds of

work equivalent to the aggregate total horses-power developed by the engines (line 97), by the product of the lineal speed of the piston in feet per minute into the net area of the pistons of the large cylinders. The quantity on line 82 was thus computed. As the equivalent back pressure against the pistons of the large cylinders must be the difference between the total pressure above zero and the indicated pressure, the quantity on line 83 is obtained by subtracting the quantity on line 80 from the quantity on line 82.

The quantities on lines 84, 85 and 86 are the percentages respectively which the equivalent indicated pressure, net pressure and back pressure are of the total equivalent pressure above zero on the pistons of the large cylinders.

Horses-power.—Lines 87 to 98 give the various horses-power developed. The quantities on lines 87, 88 and 89 are respectively the indicated, net and total horses-power developed in the small cylinder, calculated for the entire net area of the piston of that cylinder and for the pressures on lines 67, 68 and 69. The total horses-power (line 89) is for the total pressure in the small cylinder down to the zero of pressure. The horses-power developed in the small cylinder by the expanding steam alone (line 90) is calculated for the entire net area of the piston of that cylinder, for the fraction of the stroke of the piston performed after the closing of the cut-off valve and for the pressure on line 70, which is the mean pressure of the steam in the small cylinder above zero for the portion of the stroke of the piston between its end and the point in the stroke at which the cut-off valve closed.

Lines 91 and 92 give respectively the indicated and the net horses-powers developed in the large cylinders. These powers are calculated for the entire net areas of the pistons and for the pressures on lines 76 and 77. Line 93 contains the total horses-power developed by the pistons of the large cylinders, calculated for the pressure on line 78, and for the difference between the net areas of the pistons of the large and small cylinders. Only this difference of areas is to be used for this pressure, because in the case of the small cylinders, the total pressure (line 69) being taken down to the zero of pressure, covered an area on the pistons of

the large cylinders equal to the area of the pistons of the small cylinders. The total horses-power developed by the expanding steam in the large cylinders (line 94) will be, as there was no cushioning, precisely the same as the total horses-power developed.

Lines 95, 96, 97 and 98 contain the aggregate indicated, the aggregate net and the aggregate total horses-power, and the aggregate horses-power developed by the expanding steam for the entire engine—that is, these powers are the sums of the corresponding powers in both cylinders.

Weight of steam accounted for by the indicator.—The indicator accounts for the weight of steam present as such in the cylinder at any point of the stroke of its piston, and it accounts, too, for the weight of steam condensed in the cylinder to furnish the heat transmuted into the power (measured down to the zero of pressure) developed by the expanding steam after the closing of the cut-off valve. This last quantity does not appear directly on the indicator diagram, but it affects that diagram, and just to the extent that the steam is used expansively. If there was no condensation of steam in the cylinder to furnish the heat transmuted into the power developed by the expanding steam alone, the expansion steam line on the diagram would be higher than it is by an extent corresponding to this condensation. The weight of steam, therefore, which is accounted for by the indicator, is the sum of the weight which appears on the diagram, and the weight due to the condensation liberating the heat which produces the power developed by the expanding steam alone. If this sum be deducted from the weight of steam entering the small cylinder, the remainder will be the condensation effected by the metal of the cylinder. Although the portion of the indicator diagram, formed after the closing of the cut-off valve, may be, and really is, due in measure to the revaporization in whole or in part of the water of condensation effected in the cylinder before the closing of that valve, yet neither this fact nor the possibly repeated condensations and revaporizations during the expansion portion of the stroke of the piston, nor the fact that the weight of steam shown during the expansion por-

tion of the diagram is less than the weight of steam which entered the cylinder, and in variable degree, can in the least respect influence the propriety of the foregoing method of determining the weight of steam condensed during the expansion to furnish the heat producing the power of that expansion, because the steam of the revaporized water of condensation in the cylinder is used expansively, as well as the steam which has always remained as such in the cylinder. The progress of the piston and the revaporization of the water of condensation in the cylinder are simultaneous, and a particle of this water is no sooner vaporized into steam of any pressure than that steam begins and continues to expand, owing to the continuously increasing space it has to fill, created by the continuous progress of the piston.

All the steam shown on the indicator diagram after the closing of the cut-off valve being expanded steam doing work by its expansion, whatever power is created by this steam is created by the transmutation of part of the heat in it into this power, and, as an inevitable consequence, there is a corresponding liquefaction of the steam; just so much steam is stricken out of existence as steam and replaced by a relatively very small bulk of water of equal weight. The weight of steam thus liquefied can be accurately calculated from the indicator diagram. For every 789¼ foot-pounds of work done by the expanding steam on the diagram, after the closing of the cut-off valve, one Fahrenheit unit of heat disappears, so that the division of the number of foot-pounds of work done per hour by the expanding steam alone by 789¼ gives a quotient which, divided by the latent heat of steam of the mean pressure of the expanding steam, will give the number of pounds of steam liquefied per hour.

On line 99 is the number of pounds of steam accounted for per hour by the indicator in the small cylinders at the point of cutting off the steam. The bulk of this steam is the space displacement of the piston per hour up to the point at which the steam was cut off, plus the space filled per hour in the steam passages at one end of the cylinders filled at that point; the pressure is the pressure on line 63. This bulk of steam in cubic feet multi-

plied by its weight per cubic foot, must be lessened by the product of the number of cubic feet in the steam passages at the end of the cylinder filled per hour into the weight of steam per cubic foot of the pressure (line 66) of the steam filling this space when the piston began its stroke. The result is given on line 99, and no addition is to be made to it for heat consumed in the production of the power up to that point in the stroke of the piston, for this heat has already been supplied in the boiler.

On line 100 is the number of pounds of steam present per hour in the small cylinders at the end of the stroke of their pistons, calculated as in the case of the quantity on line 99, but the quantity on line 100 is not the whole weight of steam accounted for by the indicator at the end of the stroke of the pistons of the small cylinders. There must be added to it the weight of steam on line 101, which is the weight liquefied in the small cylinders to furnish the heat transmuted into the power developed by the expanding steam between the point of cutting off and the end of the stroke of the pistons, said weight being calculated as previously described. The quantity on line 102, which is the sum of the quantities on lines 100 and 101, is the weight of steam accounted for by the indicator in the small cylinders at the end of the stroke of their pistons.

On line 103 is the number of pounds of steam present per hour in the large cylinders at the end of the stroke of their pistons, calculated as in the case of the quantity on line 99; that is, the space displacement per hour of the pistons of the large cylinders in cubic feet, plus the space in cubic feet in the steam passages at one end of the cylinders filled per hour, is multiplied by the weight per cubic foot of steam of the pressure (line 73) at the end of the stroke of the pistons, and from this product is deducted the product of the space in cubic feet in the steam passages at one end of the cylinders filled per hour multiplied by the weight per cubic foot of the back pressure steam (line 75) of the pressure at the beginning of the stroke of the pistons. Now, in order to obtain the weight of steam accounted for by the indicator at the end of the stroke of the pistons of the large cylinders, there must be added to the quantity on line 103

the weight of steam liquefied in both the small and the large cylinders to furnish the heat transmuted into the horses-power developed by the expanding steam in both cylinders; that is the horses-power on line 98. Consequently, there must be calculated for the total horses-power (line 94) developed in the large cylinders—all the power developed therein being that of expanding steam—the weight of steam which would be liquefied in the large cylinders alone to furnish the heat transmuted into the total horses-power developed in them, as previously described, and then add to that weight the weight of steam on line 101, the sum of the two being the quantity on line 104, which quantity is the number of pounds of steam liquefied in both cylinders per hour to produce the entire horses-power (line 98) developed by the expanding steam in the engines. The sum of the quantities on lines 103 and 104 is the number of pounds of steam accounted for by the indicator at the end of the stroke of the pistons of the large cylinders, which sum is given on line 105. If the quantity on line 101 be deducted from the quantity on line 104, the remainder will be the number of pounds of steam liquefied in the large cylinders alone to produce the horses-power (line 94) developed therein by the expanding steam.

Previous to the closing of the cut-off valve on the small cylinders, those cylinders are in common with the boilers, and the heat which is transmuted into the power developed by the pistons of those cylinders up to that point in their stroke, is supplied by the combustion in the furnaces; hence, no liquefaction of the steam takes place in the cylinders, due to this cause, previous to the closing of the cut-off valves. But the closing of these valves puts the cylinders out of communication with the boilers, and consequently excludes this supply of heat from the furnaces, and as the development of power can only be made at the expense of heat, the necessary heat has to be taken from the steam itself, and as heat cannot be taken from saturated steam without the liquefaction of a corresponding portion of it, such liquefaction occurs, and to the extent previously described.

Difference between the weight of water vaporized in the boilers

and the weight of steam accounted for in the cylinders by the indicator.—Lines 106 to 111, both inclusive, show the difference in pounds per hour and in percentages between the weight of steam furnished by the boilers to the cylinders alone—that is, exclusive of the steam supplied to the jackets of the cylinders and the weight of steam accounted for per hour in the cylinders by the indicator—including, of course, the steam liquefaction required to produce the power developed by the expanding steam alone in the cylinders.

The number of pounds of steam supplied per hour by the boilers to the cylinders alone is given on line 53. Deducting from that quantity the quantity on line 99, there remains the quantity on line 106, showing (supposing no steam leakage by pistons or valves) that in the small cylinders, at the point of cutting off the steam, there was condensed the enormous weight of 1068.546 pounds of steam per hour, equivalent to 40.492 (line 10) per centum of the weight of steam entering these cylinders, and this, too, notwithstanding that the cylinders must have received a considerable quantity of heat from their jackets filled with steam of boiler pressure. The quantity of heat so received may be inferred from the facts that the temperature of the steam in the jackets was about 348 degrees Fahrenheit, while the mean temperature of the steam in the cylinders for both steam and exhaust strokes was about 270 degrees. The cylinder refrigeration effected in the small cylinders per hour, due to all causes, was about ($1068.546 \times 979 =$) 1046106 Fahrenheit units of heat. The heat put into the water in the boilers per hour was 2855828 units, of which the small cylinders destroyed 36.63 per centum. The inner surface of the two small cylinders exposed to the heating and cooling action, comprising sides, ends, pistons, piston rods and steam passages, was about 30 square feet, and as this surface was heated and cooled 2607 times per hour, its efficiency will be represented by ($2607 \times 30 =$) 78210, so that each virtual square foot of surface destroyed per hour *under the experimental conditions* ($\frac{1046106}{78210} =$) 13.376 Fahrenheit units of heat, but this result cannot be predicated of any other than the experimental conditions.

The refrigerating influences in the cylinder come into action almost wholly after the closing of the cut-off valve, and their effect is to lessen the temperature of the metal of the cylinder by absorption of its heat. Now when, on the return stroke, the steam enters from the boilers and continues to enter until the cut-off valve closes, the metal of the cylinder cooled from the temperature of the boiler steam during the expansion portion of the stroke of the piston, is heated to that temperature again by the entering steam which undergoes an equivalent liquefaction. This is the condensation which is measured in the cylinder by the indicator at the point of cutting off the steam.

When the pistons of the small cylinders reached the end of their stroke, the difference at that point between the weight of steam accounted for by the indicator and the weight which entered the cylinders was 798.482 pounds per hour (line 108), showing that from the point of cutting off the steam to the end of the stroke of the piston (1068.546 — 798.482 =) 270.064 pounds of water of condensation had been revaporized per hour. On the opening of the exhaust valve at the end of the stroke of the piston, this 798.482 pounds of water of condensation per hour began to revaporize, and at some point before the completion of the exhaust stroke of the pistons it had wholly revaporized, the resulting steam passing out of the small cylinders into the large ones and being there utilized for the production of power.

The entire weight of steam which entered the small cylinders enters the large ones, less probably the weight liquefied (line 101) to furnish the heat transmitted into the power developed by the expanding steam alone in the small cylinders, which passes into the large cylinders in the form of mist or excessively comminuted spray, intimately mingled with the steam.

The steam received by the large cylinders is expanded from the beginning to the end of the stroke of their pistons, the fact of there being a cut-off valve on these cylinders and of its closing when 0.515 of the stroke of their pistons was performed, not affecting the measure of the expansion; and when the pistons of the large cylinders reached the end of their steam stroke, there

remained in the cylinder 166.07 pounds of water of condensation (line 110) aganist 798.482 pounds in the small cylinders when their pistons had reached the end of their stroke. During the steam stroke of the pistons of the large cylinders there were condensed therein the above 166.07 pounds of steam, which during the exhaust stroke of those pistons revaporized, the resulting steam passing into the condenser.

That the revaporization was gradual as the pistons of the small cylinders progressed, during the expansion portion of their steam stroke, and during the exhaust stroke of the same, is shown by the continuously increasing length of the ordinates of the expansion part of the indicator diagrams taken from the small and large cylinders above the length they should have been according to the Mariotte law, or any empirical law defining an adiabatic curve for expanding steam doing work.

The fact that in both the small and the large cylinders the ordinates of the indicator diagrams are in a higher ratio than given by the Mariotte law, the increase commencing with the expansion, is largely due to the steam jackets, as without them the cylinder condensation would have been much greater and the revaporization much less during the expansion portion of the stroke of the pistons, while the revaporization during the exhaust stroke of the pistons of the large cylinders would have been much greater. There must be kept in view that the cylinder of a steam engine, whether jacketed or not, acts during a double stroke of its piston as condenser and as boiler, and, during the expansion portion of the stroke, simultaneously as both. During the non-expansion portion of the stroke the action is that of a condenser solely, and during the entire exhaust stroke the action is that of a boiler solely.

The following table shows the relation for the small cylinders alone, between the ordinates as given by the mean of all the indicator diagrams taken from these cylinders and as given by the Mariotte law: Fraction of the stroke of the pistons of the small cylinders completed when the steam was cut off, 0.290. Fraction which the steam space in the passages at one end of the small cylinders when the steam is cut off, is of the space displacement

of the pistons of the small cylinders per stroke, 0.034. Steam pressure in the small cylinders at the point of cutting off the steam, 110 pounds per square inch above zero.

Fraction of the stroke of the pistons of the small cylinders at which the ordinate of the indicator diagram is taken.	Steam pressure in pounds per square inch above zero according to the ordinates of the indicator diagram.	Steam pressure in pounds per square inch above zero according to the Mariotte law at the ordinates.
0.35	90.8	92.812
0.45	71.8	73.636
0.55	59.4	61.027
0.65	50.2	52.105
0.75	43.4	45.451
0.85	38.6	40.317
0.95	35.0	36.220

From the above table, the curves formed by connecting the free ends of the ordinates as given by the indicator diagrams and as given by the Mariotte law, would be nearly parallel, the distance between them increasing gradually and but slightly from the point of cutting off the steam to the end of the stroke of the pistons.

Economic Results.—Lines 112 to 129, both inclusive, give the cost of the indicated, the net, and the total horse-powers, in various modes of measurement. Lines 112, 113 and 114 give this cost in the number of pounds of feed water vaporized in the boilers per hour to produce them respectively; and lines 115, 116 and 117 give this cost in the number of Fahrenheit units of heat above the temperature of the feed water consumed per hour to produce them. The latter is the true cost for economic comparison with the performances of other engines.

The cost of the above horse-powers is likewise given on lines 118, 119 and 120 in the number of pounds of feed water vaporized in the boilers per hour to produce them respectively, exclusive of the number of pounds vaporized in the boilers per hour to furnish steam to the steam jackets; and lines 121, 122 and 123

give this cost in the number of Fahrenheit units of heat above the temperature of the feed water consumed per hour to produce these respective horse-powers, exclusive of the heat required to supply the steam jackets.

Lines 124, 125 and 126 give the number of pounds of crude anthracite consumed per hour in the furnaces to produce respectively the indicated net and total horse-powers developed by the engines. Lines 127, 128 and 129 give the cost of these horse-powers in pounds of what remains of the crude anthracite after deducting its refuse of ash, clinker and small unconsumed pieces falling through the grates, but including the hygroscopic water.

The fuel cost of the powers, like the duty of the engine, has but little engineering value, as it includes the economic efficiency of both boiler and coal, the skill and attention of the fireman, the rate of combustion per hour per square foot of water-heating surface, etc.

Vaporization—Lines 130 to 136, both inclusive, give the potential and economic vaporizations of the boiler with anthracite and with the combustible portion of the anthracite, under the assumptions that the water vaporized during the experiment was furnished at the temperatures of 100 and of 212 degrees Fahrenheit, and that the vaporization was effected under the standard atmospheric pressure, lines 131 and 132. These quantities are not strictly correct, for had the vaporization been performed under the atmospheric pressure and from the temperatures just stated, the results would have been greater than are given on lines 130 and 131, inasmuch as the difference between the temperature of the gases of combustion in contact with the water-heating surfaces, and the temperature of the water in contact with these same surfaces, would have been greater. The quantities on lines 132 and 133 are the quotients of the division of the quantity on line 130 by the quantities on lines 9 and 11 respectively. The quantities on lines 134 and 135 are the quotients of the division of the quantity on line 131 by the quantities on lines 9 and 11 respectively. The quantities on lines 132, 133, 134 and 135 are the economic results given by the boiler with the fuel consumed, and at its rate of consumption.

The quantity on line 136 is the quotient of the division of the quantity on line 131 by the number of square feet of water heating surface in the boilers, and by the quantity on line 1.

The difference between the assumed temperature of 212 degrees Fahrenheit for the water in the boilers under the atmospheric pressure, and the temperature of 348.29 degrees which that water actually had under the experimental pressure, is 136.29 degrees; and, assuming 1800 degrees Fahrenheit for the temperature of the gases of combustion in the furnace when generated, this difference of temperature is $(\frac{136.29 \times 100}{1800} =)$ 7.57 per centum of it; conseqently, all the vaporizations on lines 130 to 136, both inclusive, should be increased in the ratio of (1.0000 — 0.0757 =) 0.9243 to 1.0000, in order to have the vaporizations that would have existed had the pressure in the boilers been that of the atmosphere instead of that of the experiment, all other things remaining the same.

TABLE CONTAINING THE DATA AND RESULTS OF THE EXPERIMENTS.

Cylinders steam jacketed on sides and ends. Independent cut-off valves on both cylinders. Steam superheated. Neither release nor cushioning.

Date of the experiment	1st June, 1886.

TOTAL QUANTITIES.

1	Duration of the experiment in consecutive hours and minutes	12—5
2	Double strokes made by the pistons of the engines	15,751.
3	Pounds of feed water pumped into the boilers from the hot well	31,885.566
4	Estimated number of pounds of water of liquefaction passed additionally as feed water from the steam jackets of the cylinders into the boilers, taken at 286 pounds per hour, as given by another trial under the same conditions	3,455.833
5	Pounds of feed water pumped into the boilers and vaporized there from the temperature of the feed entering them	35,342.400
6	Fahrenheit units of heat put into the feed water entering the boilers	34,507,924.733
7	Per centum which the water of liquefaction from the steam jackets is of the feed water vaporized to supply the cylinders and steam jackets with steam	9.778
8	Pounds of feed water vaporized to supply the cylinders alone with steam	31,886.566
9	Pounds consumed of anthracite	3,547.

10	Pounds of refuse from the anthracite, in ash, clinker and dust	317.
11	Pounds of combustible consumed, that is, of anthracite less its refuse	3,230.
12	Per centum which the weight of refuse is of the weight of anthracite consumed	8.937
13	Cubic feet of water delivered by the pumps (coefficient of discharge 0.98)	414,144.956
14	Gallons of water delivered by the pumps (coefficient of discharge 0.98)	3,097,153.613
15	Foot pounds of work done by the engines external to them	3,299,111,854.119

ENGINES.

16	Number of double strokes made per minute by the steam pistons	21.725
17	Steam pressure in the boilers in pounds per square inch above the atmosphere	118.26
18	Steam pressure in steam pipe near cylinders in pounds per square inch above the atmosphere	113.00
19	Position of throttle valve	Partly closed.
20	Height of the barometer in inches of mercury	30.03
21	Pressure of the atmosphere in pounds per square inch above zero	14.74
22	Vacuum in the condenser in inches of mercury	28.00
23	Pressure in the condenser in pounds per square inch above zero	1.00
24	Fraction completed of the stroke of the piston of the small cylinders when the steam was cut off	0.290
25	Fraction completed of the stroke of the piston of the large cylinders when the steam was cut off	0.515
26	Number of times the steam was expanded in the small cylinders	3.241
27	Number of times the steam was expanded in the engines	13.314

PUMPS.

28	Water pressure in pounds per square inch above the atmosphere in the main, per guage at level of axes of pumps and near the latter	47.72
29	Water pressure on the pistons of the pumps in pounds per square inch above atmosphere, per indicator	50.22
30	Water back pressure in pounds per square inch below the atmosphere against the pistons of the pumps, per indicator	8.28
31	Water pressure in pounds per square inch on the pistons of the pumps, per indicator	58.50
32	Vertical height in feet from the water pressure gauge in the main to the water level in the well of the pumps	17.56

33	Pressure in pounds per square inch corresponding to the above vertical height	7.60
34	Pressure in pounds per square inch required to lift the delivering valves at the pumps and force the water through them	2.50
35	Pressure in pounds per square inch required to lift the receiving valves of the pumps and force the water through them	0.68
36	Number of indicated horses-power developed by the pistons of the pumps	148.7965
	TEMPERATURES IN DEGREES FAHRENHEIT.	
37	Of the water pumped into the main	65.12
38	Of the external air	58.77
39	Of the air in the engine room	76.24
40	Of the water discharged by the air pump into the hot well of the engines	83.60
41	Of the portion of the feed water measured by the meter when entering the heater	179.04
42	Of the portion of the feed water measured by the meter when leaving the heater	231.08
43	Number of degrees imparted to the feed water in the heater	52.04
44	Of the water in the boilers	348.29
45	Of the steam in the boilers (superheated)	370.96
46	Number of degrees that the steam was superheated in the boilers	22.67
47	Of the steam in the steam pipe near the cylinders	346.28
48	Of the gases of combustion before passing the heater	361.40
49	Of the gases of combustion after passing the heater	259.80
50	Number of degrees taken out of the gases of combustion by the water in the heater	101.60
	HEAT PER HOUR AND DUTY.	
51	Number of pounds of feed water pumped into the boilers per hour whose steam supplied the cylinders and steam jackets	2,945.2
52	Number of Fahrenheit units of heat consumed per hour above the temperature of the feed water, in the steam supplied to the cylinders and steam jackets	2,855,828.254
53	Number of pounds of feed water pumped into the boilers per hour whose steam supplied the cylinders alone	2,638.888
54	Number of Fahrenheit units of heat consumed per hour above the temperature of the feed water, in the steam supplied to the cylinders alone	2,607,749.394
55	Number of foot-pounds of external work done per 100 pounds of coal consumed, or duty of the engines	93,011,329.4
	RATE OF COMBUSTION.	
56	Number of pounds of anthracite consumed per hour	293.545

57	Number of pounds of combustible consumed per hour	267.310
58	Number of pounds of anthracite consumed per hour per square foot of grate surface	7.897
59	Number of pounds of combustible consumed per hour per square foot of grate surface	7.191
60	Fraction of a pound of anthracite consumed per hour per square foot of water heating surface	0.341
61	Fraction of a pound of combustible consumed per hour per square foot of water heating surface	0.311

STEAM PRESSURES IN THE SMALL CYLINDERS PER INDICATOR.

62	Pressure on the pistons of the small cylinders at the commencement of their stroke in pounds per square inch above zero	125.8
63	Pressure on the pistons of the small cylinders at the point of cutting off the steam in pounds per square inch above zero	110.0
64	Pressure on the pistons of the small cylinders at the end of their stroke in pounds per square inch above zero	34.0
65	Mean back pressure against the pistons of the small cylinders in pounds per square inch above zero	23.8
66	Back pressure against the pistons of the small cylinders at the commencement of their stroke in pounds per square inch above zero	26.2
67	Indicated pressure on the pistons of the small cylinders in pounds per square inch	51.6
68	Net pressure on the pistons of the small cylinders in pounds per square inch	47.05
69	Total pressure on the pistons of the small cylinders in pounds per square inch above zero	75.4
70	Total pressure on the pistons of the small cylinders of the expanding steam alone in pounds per square inch above zero	39.0

STEAM PRESSURES IN THE LARGE CYLINDERS PER INDICATOR.

71	Pressure on the pistons of the large cylinders at the commencement of their stroke in pounds per square inch above zero	27.34
72	Pressure on the pistons of the large cylinders at the point of cutting off the steam in pounds per square inch above zero	17.00
73	Pressure on the pistons of the large cylinders at the end of their stroke in pounds per square inch above zero	9.64
74	Mean back pressure against the pistons of the large cylinders in pounds per square inch above zero	1.75
75	Back pressure against the pistons of the large cylinders at the commencement of their stroke in pounds per square inch above zero	1.25
76	Indicated pressure on the pistons of the large cylinders in pounds per square inch	15.25

77	Net pressure on the pistons of the large cylinders in pounds per square inch	14.118
78	Total pressure on the pistons of the large cylinders in pounds per square inch	17.00
79	Total pressure on the pistons of the large cylinders of the expanding steam alone in pounds per square inch above zero	17.00

EQUIVALENT PRESSURES ON THE PISTONS OF THE LARGE CYLINDERS ALONE.

80	Indicated pressure in pounds per square inch that would be on the pistons of the large cylinders were the indicated pressure on the pistons of the small cylinders divided by the ratio (4.04667) of the area of the small to the area of the large pistons, and the quotient added to the experimental indicated pressure on the pistons of the large cylinders	28.026
81	Net pressure in pounds per square inch that would be on the pistons of the large cylinders were the net pressures on the pistons of the small cylinders divided by the ratio (4.04667) of the area of the small to the area of the large pistons and the quotient added to the experimental net pressure on the pistons of the large cylinders	25.745
82	Total pressure in pounds per square inch above zero that would be on the pistons of the large cylinders had the aggregate total horses-power developed by the engines been developed in those cylinders alone	31.028
83	Back pressure in pounds per square inch above zero which would have been against the pistons of the large cylinders had the power developed by the engines been developed in those cylinders alone	3.002
84	Per centum which the equaivalent indicated pressure on the pistons of the large cylinders alone is of the equivalent total pressure on them	90.324
85	Per centum which the equivalent net pressure on the pistons of the large cylinders alone is of the equivalent total pressure on them	82.973
86	Per centum which the equivalent back pressure against the pistons of the large cylinders alone is of the equivalent total pressure on them	9.675

HORSES-POWER.

87	Indicated horses-power developed in the small cylinders	73.6981
88	Net horses-power developed in the small cylinders	67.1996
89	Total horses-power developed in the small cylinders	107.6907
90	Total horses-power developed in the small cylinders by the expanding steam alone	55.7021
91	Indicated horses-power developed in the large cylinders	88.1403
92	Net horses-power developed in the large cylinders	81.5969

93	Total horses-power developed in the large cylinders	73.9744
94	Total horses-power developed in the large cylinders by the expanding steam alone	73.9744
95	Aggregate indicated horses-power developed by the engines	161.8384
96	Aggregate net horses-power developed by the engines	148.7965
97	Aggregate total horses-power developed by the engines	181.6651
98	Aggregate total horses-power developed by the expanding steam alone in all the cylinders	129.6765

WEIGHT OF STEAM ACCOUNTED FOR BY THE INDICATOR.

99	Number of pounds of steam present per hour in the small cylinders at the point of cutting off the steam, calculated from the pressure there	1,570.342
100	Number of pounds of steam present per hour in the small cylinders at the end of the stroke of their pistons, calculated from the pressure there	1,689.744
101	Number of pounds of steam condensed per hour in the small cylinders to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in those cylinders	150.662
102	Sum of the two immediately preceding quantities	1,840.406
103	Number of pounds of steam present per hour in the large cylinders at the end of the stroke of their pistons, calculated from the pressure there	2,128.942
104	Number of pounds of steam condensed per hour in the small and large cylinders to furnish the heat transmuted into the total horses-power developed by the expanding steam alone in those cylinders after the closing of the cut-off valve on the small cylinders	343.876
105	Sum of the two immediately preceding quantities	2,472.818

DIFFERENCE BETWEEN WEIGHT OF WATER VAPORIZED IN BOILERS AND WEIGHT OF STEAM ACCOUNTED FOR BY INDICATOR.

106	Difference, in pounds per hour, between the weight of steam entering the small cylinders from the boilers and the weight of steam accounted for by the indicator in those cylinders at the point of cutting off the steam	1,068.546
107	Difference, in per centum of the weight of steam entering the small cylinders from the boilers, between that weight and the weight of steam accounted for by the indicator in those cylinders at the point of cutting off the steam	40.492
108	Difference, in pounds per hour, between the weight of steam entering the small cylinders from the boilers and the weight of steam accounted for by the indicator in those cylinders at the end of the stroke of their pistons	798.482

109 Difference, in per centum of the weight of steam entering the small cylinders from the boilers, between that weight and the weight of steam accounted for by the indicator in those cylinders at the end of the stroke of their pistons..... 30.258

110 Difference, in pounds per hour, between the weight of steam entering the large cylinders from the boilers and the weight of steam accounted for by the indicator in those cylinders at the end of the stroke of their pistons..... 166.0698

111 Difference, in per centum of the weight of steam entering the large cylinders from the boilers, between that weight and the weight of steam accounted for by the indicator in those cylinders at the end of the stroke of their pistons..... 6.293

ECONOMIC RESULTS.

112 Pounds of feed water consumed per hour per indicated horse-power developed by the engines..... 18.136

113 Pounds of feed water consumed per hour per net horse-power developed by the engines..... 19.793

114 Pounds of feed water consumed per hour per total horse-power developed by the engines..... 16.212

115 Fahrenheit units of heat above the temperature of the feed water consumed per hour per indicated horse-power developed by the engines..... 17,646.166

116 Fahrenheit units of heat above the temperature of the feed water consumed per hour per net horse-power developed by the engines..... 19,192.846

117 Fahrenheit units of heat above the temperature of the feed water consumed per hour per total horse-power developed by the engines..... 15,720.293

118 Pounds weight of steam entering the small cylinders consumed per hour per indicated horse-power developed by the engines..... 16.305

119 Pounds weight of steam entering the small cylinders consumed per hour per net horse-power developed by the engines..... 17.735

120 Pounds weight of steam entering the small cylinders consumed per hour per total horse-power developed by the engines..... 14.526

121 Fahrenheit units of heat above the temperature of the feed water in the number of pounds of steam entering the small cylinders per hour per indicated horse-power developed by the engines..... 16,113.286

122 Fahrenheit units of heat above the temperature of the feed water in the number of pounds of steam entering the small cylinders per hour per net horse-power developed by the engines..... 17,525.610

123 Fahrenheit units of heat above the temperature of the feed water in the number of pounds of steam entering the small cylinders per hour per total horse-power developed by the engines 14,354.681

124 Pounds of anthracite consumed per hour per indicated horse-power developed by the engines 1.814

125 Pounds of anthracite consumed per hour per net horse-power developed by the engines 1.973

126 Pounds of anthracite consumed per hour per total horse-power developed by the engines 1.616

127 Pounds of combustible consumed per hour per indicated horse-power developed by the engines 1.651

128 Pounds of combustible consumed per hour per net horse-power developed by the engines 1.796

129 Pounds of combustible consumed per hour per total horse-power developed by the engines 1.471

VAPORIZATION.

130 Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 100 degrees Fahrenheit and vaporized under the standard atmospheric pressure 31,995.239

131 Number of pounds of water that would have been vaporized in the boiler had it been supplied at the temperature of 212 degrees Fahrenheit and vaporized under the standard atmospheric pressure 35,733.587

132 Number of pounds of water vaporized under the atmospheric pressure from the temperatures of 100 degrees Fahrenheit by one pound of anthracite 9.020

133 Number of pounds of water vaporized under the atmospheric pressure from the temperature of 100 degrees Fahrenheit by one pound of combustible 9.905

124 Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one pound of anthracite 10.074

135 Number of pounds of water vaporized under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one pound of combustible 11.063

136 Number of pounds of water vaporized per hour under the atmospheric pressure from the temperature of 212 degrees Fahrenheit by one square foot of heating surface 4.155

III.

SPEED TRIALS OF FAST SHIPS.

By Assistant Engineers Harold P. Norton and Walter M. McFarland, U. S. Navy.

In a recent number of the Journal, the writers spoke of some future remarks on the subject of "patent log" records as bearing on the speeds of fast ships. In view of the approaching trials of the *Philadelphia* and *San Francisco*, where the contract requirements relate chiefly to speed, it has seemed opportune to submit a brief review of the conditions of such trials with a view to eliciting a discussion and bringing out some valuable information on the subject.

The clause of the contracts for these vessels relative to speed is as follows:

"That the speed developed by said vessel upon said trial, under conditions prescribed by the Secretary of the Navy, shall be not less than an average of nineteen (19) knots per hour maintained for four (4) consecutive hours—the vessel to be weighted to a mean draught* of eighteen (18) feet nine (9) inches—it being mutually understood, covenanted and agreed that for every quarter knot of speed so exhibited above said guarantee the contractors shall receive a premium, over and above the contract price of the vessel, of fifty thousand dollars ($50,000); and that for every quarter knot that said vessel fails of reaching said guaranteed speed there shall be deducted from the contract price of the vessel the sum of fifty thousand dollars ($50,000)."

From this it will be seen that, in the most natural interpretation of the language of the contract, to earn the $50,000 premium, each quarter knot must be exactly completed. Nineteen and twenty-four hundredths knots means no premium, while

* This is for the *San Francisco;* the *Philadelphia's* draught is to be nineteen (19) feet two and a half (2½) inches.

nineteen and twenty-five hundredths means $50,000 to the contractor's benefit. Even if, which the language hardly warrants, the premium be paid proportionately for each fraction of a quarter knot, one hundredth of a knot means $2,000. In either case, an exceedingly small increment of speed becomes, at critical points, of great monetary value. It is, then, a matter of the highest importance that the speed should be determined with an exactness which can leave nothing to be desired, and the method to be adopted for this determination deserves the most careful study.

Four methods may be used, of varying degrees of accuracy, each having some special advantages and likewise drawbacks. These are:

1st. By readings of patent logs.

2d. By successive runs in opposite directions over a measured base, similar to the measured mile trials which now have such a vogue.

3d. By a continuous run at sea, the speed being based on the number of revolutions of the screw found necessary to give a knot in smooth water.

4th. By a continuous run at sea past a series of buoys or stations on shore, so arranged as to give the distance that the ship is likely to make, provision being made for accurate determination of the strength of tide and current at frequent intervals along the course.

1st. Assuming the patent log to give correct readings, or readings which admit of such correction that the final result is trustworthy, we should have a method which for simplicity and inexpensiveness would be admirable. At least two logs could be used, and, by suitable provision, probably several. The trial could take place anywhere, and would need very few observers. Unfortunately, however, all experience goes to show that the patent log is not accurate at any speed, and, at high speeds, is entirely untrustworthy. The readings are affected by a variety of circumstances, among which, for example, is the distance at which it is towed astern. But the greatest imperfection in the instrument is the erratic way in which the readings vary, making it impossible to apply any kind of a correction.

In 1885, the Navy Department had under consideration the formation of an auxiliary fleet of merchant vessels of good speed and adapted to conversion into commerce destroyers in time of war. The *H. F. Dimock*, a steamer running between New York and Boston, enjoyed a reputation for speed among maritime people in those ports, and it was decided to send a couple of officers to make a trip in her and report on her speed and other qualities. This trip was made on the 5th and 6th of September.

For the speed record two new patent taffrail logs were obtained from the Navigation Department of the New York Navy-Yard. Readings were taken simultaneously when passing light-houses or prominent landmarks, and, by previous arrangement, the engine counter was read at the same time. The weather was very favorable, the sea being smooth, with a light breeze about abeam for most of the run. The engines ran with fair uniformity, as will be seen by the opposite table.

A comparison of the distances between the various landmarks, as given by the two logs, shows at once that they not only differ from each other, but that each varies most peculiarly itself. This is shown in a marked way by the columns of revolutions per mile, which demonstrate most conclusively the unreliability of these means of measurement for accurate work.

On the trials of the *Baltimore* only one patent log was used. Had there been two, there might, judging from the data in this case, have been some other questions of the accuracy of the speed than have been already raised.

2d. Measured mile trials are so much in vogue and are so well understood, that details of the method need not be given. In the special cases in question, however, the fact that the trial is to last for four hours leads to a consideration of some variations and of certain points of special interest. Bases of various lengths might be chosen, and a ten-mile base has been suggested, in which case a 19-knot ship ought to be able in four hours to run over the base about six times, giving three runs in each direction, the mean of which, as usually taken for measured mile trials, would, except as to be stated in a moment, give a very fair result. There are, however, serious objections to such a long

Landmark.	Time of Passing.	Interval.	Reading of Counter.	Revolutions between observations.	Revolutions per minute in interval.	Starboard Log.		Port Log.		Revolutions per mile.	
						Reading.	Distance.	Reading.	Distance.	Stb'd Log.	Port Log.
	P. M.										
Execution Rock	7:13:00		251,290			5.45		3.02			
		35:20		2,328	65.89		6 79		7.60	342.8	306.3
Captain's Island	7:48:20		53,618			12.24		10.62			
		39:00		2,603	66.74		7.55		8.25	344.8	315.5
Eaton's Neck	8:27:20		56,221			19.79		18.87			
		1:02:53		4,182	66.50		11.93		13.18	350.6	317.3
Middle Ground	9:30:13		60,403			31.72		32.05			
		1:30:12		5,850	64.85		18.08		19.70	323.6	297.0
Faulkner's Island	11:00:25		66,253			49.80		51.75			
		1:08:35		4,850	70.71		12.70		15.75	381.9	308.0
	A. M.							67.50			
Plum Island	12:09:00		71,103			62.50			5.70	327.9	356.6
		31:00		2,033	65.58		6.20				
Race Rock	12:40:00		73,136			68.70		73.20			
		38:30		2,366	61.45		6.91		7.80	342.4	303.3
Watch Hill	1:18:30		75,502			75.61		81.00			
		1:13:30		4,885	66.46		13.61		15.25	359.0	320.3
Point Judith	2:32:00		80,387			89.22		96.25			
		1:46:00		7,038	66.40		15.93		22.20	441.8	317.0
Cuttyhunk	4:18:00		87,425			105.15		118.45			
		5:00:00		18,479	61.59		63.45		61.63	291.2	299.8
Nansett Head	9:18:00		306,904			168.60		180.08			
		1:00:35		3,911	64.55		12.20		14.52	320.6	269.4
Cape Cod Light	10:18:35		10,815			180.80		194.60			
		2:42:25		10,441	64.29		31.10		34.10	335.7	306.2
	P. M.							228.70			
Minot Ledge	1:01:00		21,256			211.90			6.42	333.1	299.8
		30:00		1,925	64.17		5.78				
Boston Light	1:31:00		23,181			217.68		235.12			

base. For these fast ships it is necessary to have very deep water, and it would be almost if not quite impossible to find a place where such a course could be suitably located. Moreover, as the strength of the tide would be by no means uniform during a period of four hours, any two runs in opposite directions would not neutralize its effect, but would leave a very appreciable error, much greater than the small amount which has already been shown to decide the bestowal of a large amount of money.

To runs over a course of a single knot these objections would not apply with much if any force. We have in Narragansett Bay, near Newport, a measured base which almost perfectly fulfills the requirements. It is thoroughly sheltered, has a minimum depth of 19 fathoms, is so close to the shore that the range poles can be perfectly seen, and there is ample room at the ends of the mile for turning. A 19-knot ship makes the two successive runs in opposite directions with so short an interval that the change in the force of the tide is scarcely appreciable, and probably not capable of determination with the instruments usually employed. With a base of this length at least twelve runs could be obtained in four hours, and the mean speed, as usually computed in these cases, will be as accurate an average as can be obtained.

It is, of course, understood that in such trials there must be no slowing down while off the course for the purpose of bottling up the steam; everything must be kept, as nearly as possible, uniform during the four hours, the only slowing being that due to movement in a curve instead of a straight line.

In this method the determination of the actual time on the course is of the most vital importance, and provision must be made for finding it accurately to within a tenth of a second. In a vessel going at nineteen and a quarter knots, an error of two-tenths of a second in the time on the course makes an error of two-hundredths of a knot. Timing watches, as usually made, read only to one-fifth of a second, and those familiar with them are aware that an error of only one graduation is exceedingly easy to make. It would seem, then, that some chronographic apparatus is absolutely necessary with this method. Such an

apparatus has been used, and a description is given in a paper read by Mr. J. H. Biles before the Institution of Naval Architects of England in 1882, and reprinted in Naval Professional Papers, No. 15, p. 87. This paper contains much valuable information on speed trials, and will repay careful reading.

Objection may be made to this method of testing the speed on account of the minute accuracy necessary in getting the time interval, and also because the curves at the ends of the mile seem to bring a much greater stress on the machinery than runs in a straight course. Those who have witnessed progressive trials in the engine room are aware of the apparent laboring of the engine while on the turns, and of the apparent increase in ease of running as the course straightens. On account of this supposed increase of stress with the accompanying greater liability to heating of journals, the contractors might not be willing to have the trials made in this way when everything is already under tremendous tension.

Another point needing consideration, if it be not called an objection, is the question of inertia. After the speed is once established and everything working uniformly, there is no expenditure of power necessary to maintain the *vis viva*, all going to various forms of friction. On the turns, however, the speed diminishes, and consequently the kinetic energy. To restore this to the amount required for the previous speed will take some little time and space enough on a straight course before reaching the beginning of the base. It would be well, before any ships are tried by this method, to make some experiments which would at least furnish a guide as to what the length of this straight course before beginning the mile should be. Some form of tachometer which would accurately register the speed of rotation would be a great assistance in this matter.

3d. A memorandum in regard to this method has been submitted to the Navy Department by the Engineer-in-Chief, and by his permission the following details are given.

It is necessary to determine the number of revolutions per minute and per mile for various speeds in smooth water by runs in opposite directions over a measured base. For the reasons

already given, this course should be a knot. The measured mile near Newport would answer for vessels built on the Atlantic coast, and several courses can be laid out in Santa Barbara Channel for the Pacific, and a course can be laid out in San Francisco Bay, with a minimum depth of 20 fathoms, though it would interfere with ferryboats and is subject at times to a very strong current. This latter course is between San Francisco and Angel Island. In any case, there must be a minimum depth of at least 19 fathoms.

Sets of runs at 14, 16 and 18 knots, and at the highest the contractors are willing to undertake on a "preliminary" trial, should be made; at least four, and preferably six, at each speed. On these trials the revolutions must be determined with the greatest accuracy, and other precautions taken as given under 2d. Having carefully obtained the data for these speeds, curves showing revolutions per minute and revolutions per mile, with the speeds for abscissae, should be laid down on a large scale, so that the speed to hundredths of a knot can be determined.

In laying out these curves, the direction which the spline or batten must take in order to give the curve will enable the curve to be extended with a close approach to accuracy beyond the highest point determined, and thus give the necessary figures for ascertaining the speed when, on the trial, the vessel is pushed to the utmost.

The official trial could take place anywhere, the only condition being smooth sea, and wind not exceeding (say) a force of 4 on the Beaufort scale. At least two counters should be attached to each shaft, and these of the very best pattern. A suitable system of signals having been arranged and practised, the trial can begin. Counter readings should be taken simultaneously at least every ten minutes during the trial, and, with the concurrence of the contractors, this should be continued for thirty or sixty minutes beyond the four hours required by the contract, in order that if any misunderstanding about the commencement of the trial should occur, it can be adjusted from the record. It would be well to have the counter readings also taken at ten-minute intervals before the probable commencement of the trial.

The four-hour period having been determined, and the total number of revolutions made good during that time, the average revolutions per minute is obtained at once, and from them the number per mile and the speed.

As the ships are all fitted with twin screws, it will be necessary to take the mean of the revolutions of the two shafts in laying out the curves and in determining the speeds on trial. There is no reason *a priori* against the accuracy of this course, and it is the only practicable method in twin-screw ships.

The great advantage of this method is its simplicity and cheapness, and the ease of carrying it out anywhere. It obviates all the trouble and expense of establishing special courses with their accompanying necessities for accurate tidal and current observations. It is true that it involves runs over the measured mile, and thus seems to add to the second method its own. It must be noted, however, that there is an essential difference. This method gives the contractor his choice of a suitable speed without risking hot bearings, while the second method compels him to take this risk, as he must push everything to the last point of endurance in order to secure the premium.

4th. The last method is also a sea trial, but from its limitations must take place near the shore, or at least in water that is not too deep for the location of buoys. A paper devoted to this method and a selection of suitable locations has been submitted to the Navy Department by Lieut. Taussig, who shows that there are several suitable courses in Long Island Sound and to the south of Long Island, and also in Santa Barbara Channel.

The course is to be established by suitable beacons on shore, if the necessary depth of water will permit a sufficient proximity, or buoys where it will not. These are to be placed at proper distances, so as to make the whole course from 80 to 100 miles long. Provision is to be made, by tugs or other small vessels stationed along the course, with competent observers and apparatus, for the accurate determination of the strength of the tide and currents at various points. These observations should include currents below the surface, if any exist, or if their strength is different from that on the surface. Unless the buoys are very

close together, it will be necessary as the end of the four-hour period approaches to take sextant angles of the beacons or buoys, so that the position can be accurately plotted from time to time, and be readily determined when the end of the trial occurs.

After the trial the distance as made good by observation is to be corrected for the force of tide and current and the average speed then computed.

It will be seen that the chief advantage of this method lies in the minimizing of the effect of small errors of time or distance by making the course a long one. It would probably be desirable, as in the last method, to continue the observations for a short time after the termination of the four-hour period.

The objections to the method are the first cost of establishing the course and of keeping the vessels or tugs necessary for tidal and current observations, as well as the large number of observers necessary.

There may also be a question in regard to the degree of accuracy obtainable in the measurement of the strength of the tide and current. By the use of good current meters, which have been carefully standardized, however, the accuracy of this factor can be secured to less than a hundredth of a knot; still it would be somewhat expensive to provide many such meters.

It appears to the writers that, on the whole, the third method is the one likely to give most satisfaction both to the Department and the contractors. It is worth while, then, to consider the degree of accuracy which is attainable by this method. Let us consider the case of a vessel where the screw makes 120 revolutions per minute for 20 knots. This ship would run the mile in three minutes, and, as ordinary counters read only complete revolutions, there might be an error of one in the total number of revolutions in that period, or 0.33 per minute. As 6 revolutions give a knot, this would give an error of 0.056 knots, which is more than twice that due to an error of one graduation in a stop-watch reading to one-fifth second. In the four-hour run, there should not be more than the same error of one revolution, which would here be only $\frac{1}{240}$ revolution per minute, or an inappreciable amount. The error in the first case, however, is too

great altogether, and shows that in this method an instrument capable of more delicate reading than the usual revolution counter must be employed. It will probably not be difficult or expensive to construct a device of this kind reading to one-tenth revolution. By arranging this to give a record in conjunction with a chronograph, any desired degree of accuracy could be secured. It must be noted, in this connection, that the second or measured mile method avoids this difficulty, as the only observations requiring delicacy of measurement are those for the time interval, which must, in the third method, be also taken with the same precision.

The writers trust that those members of the Society who have had experience in trial trips will give it the benefit of their observations, and thus add greatly to the value of this paper.

Discussion.

Mr. Charles E. Emery, Assistant Engineer U. S. Navy, 1861–69.—From the statements made in the paper it would appear that the general method of making the trials of the *Philadelphia* and *San Francisco* must be settled largely by the exact words of the contract, or such construction of the same as may be made in advance by the Secretary of the Navy.

Strict construction would require a run of four consecutive hours "straight away" for the reason that any attempt to return over a particular course would slow the vessel in making the turn, the engines would run correspondingly slow and the steam thereby be "bottled," even though the throttle valves and cut-offs were maintained in constant position. The contractors could not afford to have this reduced speed included, and if rejected the Government would not get the result of a straight away run on account of the bottling at the turn or, what amounts to the same thing, the chance to fix up the fires during such interval. With these particular provisions, therefore, it would seem necessary to make the run in one direction over a course of sufficient length and to determine the speeds as closely as practicable by the several methods adapted to this course of action, or, say, 1st. By the use of a patent log; and, 2d. By observation of buoys in connection with floats to to show the currents. The revolutions would of course be observed, but more as a check in connection with observations of the wind and sea.

Necessarily, preliminary runs would be made back and forth over a

short course, to determine points in a curve showing the relation between the power and the speed for a given draught of water, and these, or supplementary runs, should be made over a course of sufficient length to enable the rates of the patent logs provided for use on the contract trial to be also determined at different speeds *of the same vessel.* Necessarily the patent logs, like propellers, have a certain amount of slip, and each one has a different rate and a peculiar law of variation, which, however, is substantially constant for the same conditions. In the use of buoys, or land marks, the services of other vessels are required, not for extended observations of the currents, but to ascertain the currents during an interval including the time of trial. The Coast Survey method of using submerged buoys with tell-tales sufficiently small to be uninfluenced practically by the wind will answer every purpose, provided there are vessels enough to make the necessary observations at several points along the course. It must not be expected that absolute accuracy in a scientific sense will be secured. The practical question is to ascertain the speed with such accuracy that the officers in charge will conscientiously feel that neither the interests of the Government or the contractor have suffered, so far as known, and such methods be adopted that the contractor will be satisfied with the fairness of the trial, thereby settling all questions at once. It is a practice with old British civil engineers not to allow the assistant engineers to run over a line the second time, thereby preventing conflict of evidence. This way of avoiding difficulty, if not recommended, has its advantages, as the common sense view of the object to be accomplished should always be uppermost.

Instruments of observation and methods of investigation so elaborate that none but the promoter can properly use them should not be permitted. It is better to use apparatus and methods familiar to all and to make a run long enough to average all errors, so as, if possible, to avoid the complications due to averaging a series of trials. The nearer a trial of this kind can by a long preliminary run be brought down to the conditions of average practice, the more satisfactory it will be. Those of us who had experience with our seven and eight-knot vessels during the war, or even the phenomenal ones of eleven or twelve knots would, if the time since were a blank, be surprised at guaranteed speeds of nineteen knots per hour, and a little excitement and elaboration of methods be warranted; but a single trip on an ocean liner like the *City of Paris*, averaging twenty-one knots per hour, from shore to shore, forces one soon to realize that these speeds must be made with everything working under substantially average conditions and with as little worry and ex-

citement as with our former slow vessels, and, while elaboration of methods advances truth and leads to a higher standard of marine engineering, the general fact remains that it is best, so far as possible, to adhere to well known methods carried out in well known ways, so that the results can be known promptly and all join in testifying to their substantial accuracy.

Assistant Engineer W. D. Weaver, U. S. Navy.—About five years ago I submitted to the Bureau of Steam Engineering a design for indicating the speed of a shaft and of a ship, the principle of which could be applied to a counter, such as is suggested in this paper.

The arrangement was primarily intended as a revolution indicator, and was to have any desired number of registers attached and distributed about the ship.

On the shaft was to be secured a number of projections by means of which electrical contacts were to be obtained, and, through a clock working a commutator, the current passed to the registers, and there moved a hand showing the number of revolutions, or rather, the rate at that time. By pressing a button at the register the circuit was established, the hand moved along the dial for several seconds, until it reached the number corresponding to the rate of the revolutions per minute, when the circuit was automatically opened, the button sprung back, and the register was ready for another indication. The hand remained on the number until the button was pressed for a new reading, when it sprung back to zero and started ahead again.

At the experimental tank of the Denny Bros., at Dumbarton, I had seen a small propeller used to indicate speed in water by means of electrical contacts, and if it had been found practicable to rig a screw forward of the stem, it was my intention to have the same register, upon touching another button, to indicate the speed of the ship.

If the same principle were applied to a counter, the revolutions could be obtained to a greater accuracy than a tenth, as the electrical arrangement, which I have found to be similar to that used on a Gold and Stock "ticker," is very sensitive.

Another means of obtaining the revolutions for several minutes to a great degree of exactitude is as follows:—

An arrangement similar to a gas meter can be connected to the engine shaft, having pointers on the arbors (which are connected by a train of wheels), and giving readings to a fraction of a revolution, units, tens and hundreds respectively; behind each pointer will be a dial, the one on the fractional arbor being as large as practicable. Now, the

problem is to determine the positions of these pointers exactly at the beginning and the end of the trial course, and as the positions on the last two dials, the tens and hundreds, can be easily read at any time, we need only confine our attention to the first two, those reading to fractions and units.

By covering these dials with sensitized or smoked paper, and establishing a momentary electric current at the proper instant, a mark may be made establishing the position of the pointers at that instant, which on the unit dial can be read exactly, and on the other afterwards measured to the degree of exactitude required, there being a sufficient interval between the time of entering and leaving the course to place signs opposite the initial marks to distinguish them from the final ones.

On the sensitized paper the marks may be made by a spark from a projection on the pointers, or by a current passing through a light spring trailing on the same; on the smoked paper by means of a small magnet, well balanced on the arbor, a trailing point may be made to depart from its circular path for an instant and thus fix the required position.

To avoid unnecessary wear, the apparatus may be connected a few moments before the range is reached by means of a frictional coupling, and a bell made to give warning just before the contact is made, that a lookout may be kept in order that the first positions on the paper may be properly marked. To secure greater accuracy in marking time, the time-marker should be apprized when to work by a pricking point actuated by a magnet, whose circuit is established by the same motion as those of the two dials.

The necessity to which the writers refer of permitting the ship to attain her full speed before entering on the measured course is an important one, and the distance to be allowed for this should be found by experiment or calculation, where the course is a short one. In a paper, "Some Problems in Propulsion" which I contributed to No. 1, of this Journal, is given a formula and its deduction, by means of which this distance can be approximately calculated; in the case of the *Lepanto*, the distance to be allowed, while the speed is increasing from 16 to 17.9 knots, was found by calculation to be almost two miles.

If indicator cards are taken before uniform speed has been attained, and the resulting horse-power reduced to a mean number of revolutions, the final horse-power will be too large, and may be made very much so. While this remark has nothing to do with the subject under discussion, it is well that attention be called to the point, on account of the implied possibility of "jockeying" on horse-power trials.

PASSED ASSISTANT ENGINEER A. M. MATTICE, U. S. NAVY.—I fully agree with Messrs. Norton and McFarland that "patent" or taffrail logs are useless for the purpose of measuring the speed of fast war ships on their four-hour trials, where such extreme care is necessary to ascertain the speed to the closest degree. The discrepancies of the *Dimock's* logs teach a very instructive lesson." I, too, have noticed some very peculiar antics on the part of such logs. With a vessel running in very smooth deep water, and with no wind, the revolutions of the log should be proportional to those of the propeller, yet under these circumstances I have noticed discrepancies so great as to condemn the log for purposes like those here in question. It might be said, at first thought, that the propeller is as liable to be wrong as the log, as they are both screws acting in a liquid medium. But the fundamental difference between the two is that the propeller *as a recording instrument* is not affected by the friction of its connecting mechanism, while the log is so affected, sometimes to a great extent. If the recording mechanism of a log had a constant friction, then that part of the log error which is due to such friction would be constant; but if the friction is doubled from any cause, then the log's error, as far as it is due to that friction, is doubled. That this friction is sometimes very great, comparatively, can easily be seen by watching one of these logs in operation. The recording mechanism will sometimes remain inactive for several seconds, when it will suddenly start up with great speed and record with greater or less accuracy the accumulated twists of the line. As the only work done by the log screw is to overcome the friction of the line in the water, the stiffness of the line, and the friction of the mechanism, it is readily seen that this friction should be as constant as possible. There is no doubt that the "patent" logs can be much improved in this respect, as they are in general very inferior pieces of mechanism; but however perfect they may be made the fact remains that there is a continual frictional error of unknown amount."

That the system of successive runs over a short measured base gives delusive results is almost self evident. I am not sure, however, but that it might be well, as a matter of policy, to make one set of speed trials in this way so that the results might be compared with foreign trials made in the same manner, and then make a continuous trial for our own information. The former results, still following the practice of some of our foreign friends, could be given to the newspapers, while the latter could be kept to ourselves. What I mean is that it is not fair to rate our ships with an honest four-hours continuous speed and then

compare this speed with "successive measured mile" speeds of foreign ships, as some people are fond of doing. One very important point in connection with such trials has not been mentioned by the writers, which is as follows: In late marine designs, in order to keep down the machinery weights, the valve motion is usually arranged to give a very late cut-off, and full gear is used on the speed trials. Now suppose, as often happens, that the piston displacement, while on the measured mile is just slightly in excess of the evaporative capacity of the boilers. The steam pressure falls slowly, but when the vessel comes to the turn and the helm is put over, the resistance increases, and as the valve gear can follow no further the engines necessarily slow down and the steam pressure rises. In few words, if the engines can just work off the steam on a straight course, they cannot work it off at the turns. The consequence is that, although by the terms of the rules for the trial the steam must not be "bottled up" at the turns, that is exactly what happens, and the vessel starts on the next mile with its second wind, so to speak.

Now, although I agree with the writers that the third method mentioned is the preferable, I would, for the present at least, combine with it the fourth, simply that the one method may check the results of the other. This would undoubtedly remove any apprehensions as to the accuracy of the third method, which might subsequently be used alone. One very essential precaution to be taken in calibrating the propeller on the measured mile is to run the vessel over the mile at various displacements, as the quantity of coal used on the four-hour trial will probably in some cases change the displacement enough to make an appreciable difference in the slip of the propeller.

As to any error arising from the inability of reading the ordinary engine counters to less than one revolution, there need be no apprehension, as all, or nearly all, the engines of the new vessels are required by the specifications to be supplied with rotary counters instead of the reciprocating ones ordinarily furnished. By making the unit hands of these of sufficient length any desired degree of accuracy may be obtained. One of these counters for each propeller could easily be combined with a chronograph for use on the measured mile as follows: Let the unit arbor of the counter be connected with the driving gear by a pawl and ratchet which can be manipulated by an electro-magnet. Arrange a chronograph barrel in any of the usual ways to be driven by either an electric or hydraulic motor with, say, every tenth second registered by chronometer. The counter being out of gear, set the

hands to zero. Let the observer on deck operate a circuit closer at the instant the vessel enters the measured mile. This circuit would, by means of simple mechanism, begin to trace a line on the chronograph barrel and at the same time throw the counter into gear. At the end of the mile the breaking of the contact would end the chronographic line and would at the same instant stop the counter. (This may be similar to that described by Mr. Biles, but I have not his paper at hand, so give the above for what it may be worth.)

As to counter readings during the four-hour trial, the following would eliminate personal errors, and would give a permanent record of each reading: Let the counter be enclosed in a dark box containing a camera. At, say, every quarter hour let the chronometer make an electric contact which will cause a flash light exposure to record the exact position of the counter hands. A fresh plate can then be inserted and the previous negative developed.

I do not know but that "force 4" may be too much of a permissible wind for a trial where the speed is measured by the propeller, and I think that this might be cut down. Another precaution to be taken is to run the vessel over the measured mile again at the end of the trial, as it might happen in some cases that the propeller blades had sprung enough to affect the result.

Now, the results obtained by observations of the revolutions could be easily checked by a very simple device, if that device were at hand to be used. I refer to an instrument for indicating the actual thrust of the propeller. Such a device, as I mentioned in a late discussion, was included in the original design of the *Yorktown's* machinery, but was, for reasons not necessary to mention, omitted from the contractor's designs. This was to have been constructed on the principle of the Emery weighing machines, with provisions for eliminating the effects of friction of the shaft in the stern tube. By its use we could at any moment ascertain, to within a very small error, the actual propulsive force. If a vessel is supplied with such an apparatus, in going over the measured mile we can ascertain just what force is required to propel her at a given speed and given displacement in smooth water with no disturbing wind. Now we can take the vessel outside and run the four hours' trial without fear of any disturbance from a slight wind or sea, as far as its effect in retarding or assisting the vessel's speed is concerned. Suppose that we start off on the trial all right, but that a head wind springs up. Ordinarily we would have to abandon the trial, but if we read our thrust dynamometer and our engine counter we can as-

certain by a simple multiplication how much work the propeller is doing. Now if this is the work which the measured mile trial has demonstrated as necessary for a certain speed we should credit the contractor with that speed, whether or not the ship, under the influence of the disturbing wind, were making it or not. In the same way the effect of a stern wind in assisting the speed could be allowed for and deducted. In short, if the measured mile trial shows that a certain amount of work will give a certain speed, we should credit the vessel with that speed for four hours, if the propeller does the work, notwithstanding the effect of wind or sea in retarding or assisting. In cases of very close cylinder capacity this might work slightly against the contractor's interest, as a head wind, although slight, might knock the cylinder capacity down below that of the boilers, so that the measured thrust work would not be as great as might have been with no wind. This method of measurement must not be confounded with horse-power trials, as it takes into account the efficiency of the propeller and eliminates the effect of friction. A dynamometer of this sort is simple and is easily applied to a thrust bearing if worked out in the original design. No one who has witnessed the work of an Emery testing machine can doubt the accuracy with which it can be made to indicate any given force, or its applicability to thrust dynamometers as here suggested.

PASSED ASSISTANT ENGINEER E. T. WARBURTON, U. S. NAVY.—The presentation of methods of measuring the speed of fast ships is timely, in view of the fact that two vessels, which are expected to make 19 knots or more per hour, will soon have their trial trips, and the necessity for an accurate determination of the speed is apparent.

But one instrument was used on the trial of the *Charleston* for ascertaining the speed, a Walker Cherub log, with balance wheel. This log had been in use on the S. S. *Corona* at speeds of about 13 knots, and had also been tested by the S. S. *Queen of the Pacific*, at a maximum speed between 15 and 16 knots. On the Pacific coast, where fogs prevail, great reliance is placed on the Walker Cherub log, which is considered the best instrument of its class; vessels frequently make runs between two ports in a fog, and the recorded distance is found to be exact, or nearly so. Still, for higher speeds, without careful tests, the patent log could not be depended upon solely, though it might be well to use it in connection with Method 4, particularly on the Pacific coast, where, before the end of the run, fog might obscure the buoys or beacons.

The distance run by the *Charleston*, as recorded by the Walker log, in six hours was 109 knots; in the last four hours of this continuous run,

being the period of the official trial, the distance recorded was 72.75 knots, average 18.18 knots per hour, and a maximum of 18.75 knots. Previously to the trial, careful observations of points on shore gave a speed of 19.6 knots per hour, and at other times 19 knots; it is the opinion of some who were on the trial that the speed recorded by the patent log was too low, and that when the *Charleston* has her progressive speed trials she will be given a higher rate. This throws doubt upon the accuracy of this particular patent log for high speeds, while it was probably correct for moderate speeds.

The third method, if carefully conducted, the observations and measurements made with instruments of precision, thus eliminating all chance of error, would undoubtedly give the true result.

In the *Charleston*, while the total number of revolutions of the engines recorded by two observers, from readings from two counters, agreed, there were sometimes errors in the observations for ten-minute intervals; these personal errors could be readily overcome by the use of automatically recording instruments.

There would be an advantage to the Government in the adoption of this method, that there would be no necessity for progressive speed trials after the acceptance of the ship, as the various speeds and powers would be determined by the Trial Boards on the preliminary runs, and the full performance of the vessel known at an early date, without waiting for the training of the crew.

A course of one knot or more could be laid off in San Francisco Bay, between Angel Island and Red Rock, with reasonable security from interference by passing vessels; and it is probable the contractors would be willing to speed the vessel to within a very small fraction of the maximum point.

Where the full course could not be laid off, this method, No. 3, is the most practicable and exact; but on the Pacific Coast, where Santa Barbara Channel offers a long stretch of smooth, sheltered water, its merits should be considered in connection with Method No. 4. In Santa Barbara Channel the force of the current is not subject to the changes and fluctuations of a more open course, but is practically constant, and the number of buoys and stations for current observations would be a minimum.

An official of the Union Iron Works, whose opinion may reach the Society through another source, suggests the placing of four schooners in an eighty-knot course. This would give a sufficient number of current observing stations for Santa Barbara Channel, though more could

be added, and signals on the mastheads would enable a straight course to be steered. This method would do away with the preliminary speed runs over the measured mile, and it is doubtful whether the cost of buoying the full course, and the observations for currents, would exceed the cost of the progressive speed trials of Method No. 3; and in my opinion it is preferable for the Pacific Coast.

I do not agree with the writers of the admirable paper under discussion, that a full quarter knot over 19 knots must be attained before the first increment of premium is earned; if this were so, no penalty could be exacted if the vessel made less than a quarter knot below 19 knots.

In the completed ships, already tried and accepted, a fraction of a horse-power earned its premium, and it seems but just that this ruling should apply to the speed requirement.

MR. W. R. ECKART, MEMBER.—On this matter of speed trials, I am in favor of progressive trials and constructed speed curves.

With regard to that portion of the paper that refers to the chronograph records, I may say that I have an instrument, more than three-fourths finished, that is intended to be used on the trial trip of the *San Francisco*, if it meets the approval of the Navy Department and the contractors. It consists of the chronograph that I used on the *Comstock* for measuring the velocities of pistons of steam engines, pumps and pump rods, when I was conducting some experiments for the Government. (A cut and description of the instrument, as originally made, is given in a paper I read before the American Society of Mechanical Engineers, published in Vol. III of their Proceedings.) It is defective for use on board ship, but with the modifications and change of driving power, and the addition of a larger drum, the difficulties have been overcome. I propose to permanently record the revolutions of each engine, so that they can be read to at least the tenth part of each revolution, and the time recorded to the 1–500th part of a second, if necessary. Alongside of these records will be recorded each knot and fraction of a knot from any taffrail log that is used. Further, the exact time of passing any mark in the speed course can be indicated and recorded by one or *more* observers. As all these records are made on one sheet of prepared paper, which becomes indelibly fixed by being covered with a slight coating of varnish, as described in the paper above referred to, the Navy Department would always have the record before them that in no way can be "cooked" after once made. I have spent a great deal of time and money on the instrument, and I consider that every detail has been carefully thought over, and it has been my intention to ask for permission from the Depart-

ment and the contractors to use the machine for recording results from the *San Francisco* trials. I consider that the instrument is far superior to the one used by Mr. J. Harvard Biles, and mentioned in his paper. When the instrument is completed, I propose to send the Bureau of Steam Engineering a photograph of the same, and would be pleased to have their criticisms, and any suggestions as to improvements.

Further, I can state that a good many years ago I made a great many experiments with a Berthon log, following Berthon's instructions, as read before the British Association, Section G, and also described by Vaughan Pendred in *Engineer* of Dec. 17th, 1869, and by Prof. Rankine, Jan. 5th, 1870. My experiments led me to believe that when properly constructed, handled and located, it is a valuable instrument, and I would very much like permission from the Department to design and build one for use on the *San Francisco*. The expense would be slight, and the information gained, I consider, would be valuable, and having once corrected the log over the speed course proposed, it could ever afterwards be used with reliance for that ship. My reference to this log may meet with criticisms from those who have not carefully considered the subject, or who have not experimented with one of them, but I am speaking from experience that I gained from the construction and use of one that I placed and experimented with on the Steam Launch No. 4, referred to in Isherwood's Experiments on Screw Propellers, made at Mare Island Navy Yard.

MR. G. W. DICKIE, MANAGER UNION IRON WORKS, SAN FRANCISCO.—Of the four methods given in the paper, accurate results could only be expected from the 3d or 4th method, although I would not accept the table of log errors given as representing the errors usually found in good logs. We found on the *Charleston* that while the logs supplied by the Navy Department were very unreliable, the log borrowed from the Pacific Coast Steamship Co. was very correct. Where so much running is done in foggy weather, as is the case on this coast, great attention is given to the correctness of the log used, and those used on the fast coast steamers here will not be one mile in one hundred in error; but this would, of course, be too much for the purpose in view. Successive measured mile runs would not be satisfactory to either party on account of what may occur between the runs.

I would be in favor of the 3d method proposed, as it would be easier to carry out on the final trial than the 4th method, but it requires that the preliminary runs shall be conducted with great care, and I doubt whether a one mile base is long enough to establish the mile revolutions

at various speeds. The error of the fraction of a second, and the fraction of a revolution, if both errors are in the same direction, and extended into a run of four hours duration, might be a very important factor in the final result. The writers of the paper appear to think that the error of revolutions in the one mile would only be the same error on the four hours run, but the revolutions shown as required for the one mile run is the divisor for the total revolutions of the four hours, so that the error is multiplied by the number of miles run. In the case of the ships under consideration, I would propose, if this method were used, to have the test base five miles long and buoyed at one mile intervals. During the time the runs are being made, have a free buoy liberated from each of the fixed buoys; these free buoys to consist of a piece of timber, weighted on one end to stand vertical in the water, the length of the submerged part to be 18 feet 9 inches, or the mean draught of the vessel. These would determine the amount of current, if any, and its direction. There should be five runs made each way over this base, say at 10, 12, 14, 16 and 18 knots speed. In this case an error of the fraction of a revolution, or the fraction of a second, would be a very small factor in the final result, while the five speed intervals would enable a very correct speed and revolution curve to be constructed.

The 4th method would be the best were it not for the difficulties that would occur in measuring tide-effects over so long a base, and while it might be successful in Santa Barbara Channel, where there is little or no tide, and that a constant for months, yet it might not be possible to find on the other side a similar condition as to tide.

I must also point out an important matter in using the 3d method. If five sets of revolutions were obtained for speeds of 10, 12, 14, 16 and 18 knots, and the extension of the curve thus produced could give the revolutions required for higher speeds, the correctness of this curve determines the value of the method. Now, should the point at 16 knots be placed a little low by a slight error in time or revolutions, and the point at 18 be placed a little high from the same cause, the result would be that the curve produced would require a higher number of revolutions than the true amount for the speeds above 18, while if 16 were a shade high, and 18 a shade low, a less number of revolutions than the true amount would be required for the speeds above 18. The curve produced is simply the increase of slip that follows from an increase of power applied to the propeller, and may not at high speeds produce a true curve. With all the possibilities of error, it looks like taking chances as to whose side the errors will favor.

With the 4th method, and Santa Barbara Channel to make the trial in, there need be no question of error. Let five schooners be chartered and moored stem and stern at intervals of 19 knots apart, with a pole fixed on each, marking the correct distance. The current, if any, could be found at each schooner by the method already given. The exact revolutions for each 19 miles could be taken as a measure for any distance made beyond the course, or short of it. This would be a trial that the whole country could understand, as there would be no constructed curves, needing an expert to understand, to obscure the actual result.

MR. ROBERT FORSYTH, OF UNION IRON WORKS, SAN FRANCISCO.—In considering the different plans of obtaining a correct record of speed in fast vessels, the ordinary log, as used in navigating, is not to be considered as at all reliable.

To rate the speed by recording the revolutions of engines, after obtaining by trial the speed of the vessel at different rates over a measured distance, is more likely to be correct, if the distance used as a base is long enough and correctly measured, and pains taken to get the engine revolutions and time noted to fractional parts.

For the *San Francisco* I would suggest that the speed trial be made over a distance of 76 miles in smooth water, say Santa Barbara Channel. That this distance be marked by mooring five schooners equal distances apart, noting correctly the time, revolutions of engines, wind, tide, change of course, if any, during the run. And if the vessel overruns the 76 miles distance, the remaining distance can be computed from the engine revolutions, at the same rate obtained by observation on the measured distances. Two officers could be stationed on each schooner, and with tide meters, or loose buoy, note the rate of tide when vessel passed. They could also note the wind, and the ship could be permitted to drift (afterwards) before the same rate of wind to obtain the effect the wind had for or against her speed.

The greatest care would require to be taken to get the *fractional* parts of the revolutions correctly noted, also the *fractional* parts of the *seconds* of time when passing the measured space.

Special electrical apparatus would be required in duplicate for this purpose, and the observers well drilled to their duties beforehand.

PASSED ASSISTANT ENGINEER G. W. BAIRD, U. S. NAVY.—The subject under discussion is one that has always been, to me, one of extraordinary interest, but the present demand for accuracy has, I must admit, far surpassed my expectation. It is, of course, essential to obtain exact

quantities, not only for the benefit of the Government and the contractor, but for the sake of truth and science. Experiments on so large a scale can only be afforded by a Government, and it is to Government engineers that the mechanical and mercantile world naturally turns for plain, hard, unvarnished facts.

We have heard the steamboat men's stories concerning the marvelous speed of the *Francis Skiddy*, of the *Armenia* and the *Mary Powell*, which, it was currently reported, reached the great speed of 27 miles an hour on the Hudson River. But these statements are now regarded as steamboat men's stories; statements uttered in good faith, and probably with much truth, but as no mention was made of tide, current, wind, or revolutions the real speed through the water has ever been doubted by other steamboat men.

In our experiments on the screw propeller at Mare Island, where we used the best instruments the state of the art then afforded, and where we employed every possible check on our work, we found that a speed gauge, designed by Mr. W. R. Eckart, late Second Assistant Engineer, U. S. N., agreed, in a distance of two miles, with the shore marks which had been carefully laid off by the civil engineer, Mr. Calvin Brown, within four-tenths of one per cent. This gauge was essentially a velocimeter, but it would not be difficult to so improve it as to insure a continuous line of record. This gauge I believe to be much more accurate than any of the "logs" now in use, for the reason that the slip of the "logs" not only varies with their velocity through the water, but with the varying conditions of the sea, and the direction of the wind. Fortunately, the divisions on the scale of this gauge are larger at the higher speeds. The construction of this admirable gauge is described in the report of the Engineer in Chief of the Navy for the year 1874, page 19. Briefly, it is Pitot's tube with Berthon's attachment. This attachment is the addition of a tube, and the angles of the orifices, as determined by Berthon. The gauge was designed by Mr. Eckart, and built in the steam engineering shops at Mare Island.

I afterwards adapted a simple form of the Pitot tube to the *Albatross*, but was detached from that ship before I had it fairly completed. I think, however, that I placed it too near the keel to get satisfactory results.

Very difficult observations to make, in these coming trials, will be those on deck, in determining the ends of the bases over which the ships are run, for if ranges are used on deck the deflection of one degree from the true course of the ship would cause considerable error.

The speed of a ship is usually determined by running over a measured course; but I have never heard of a credit being given for bad steering. Stand upon the stern of certain ships, in a smooth sea, and look at their wakes, and you will readily see, from the zig-zag course, that they travel a greater distance than the plain, recorded distance between posts. The same thing probably takes place in a speed trial. We therefore require a graphic and an automatic diagram of the course. Another, which would record the exact course of the vessel the instant she crossed the bases; another, which would record to tenths of revolutions of the engines per minute.

In the present state of the arts, there would be, I imagine, no great difficulty in obtaining instruments to determine these quantities to a tolerable degree of accuracy. The remaining interrupting force would be that of the currents through which the vessel passed. Lieutenant Pillsbury's instrument might probably be employed for this purpose, at several points in the course, and a mean taken, which would afford an additional correction.

ASSISTANT ENGINEER MCFARLAND.—I wish to say, for the writers of the paper, that we are much pleased with the interesting and valuable discussion that has been elicited. With regard to patent logs, I would say that the one used on both of the *Baltimore's* trials was a Walker Cherub log, like that used on the *Charleston*, but an inspection of the table of speeds and engine revolutions for the second trial (given elsewhere in this number) shows that it gives very inaccurate results. The tabulation of maximum data brings out this discrepancy in a marked way, where about four revolutions and 900 I. H. P. more give, per log, a knot less speed. This, of course, is absurd, and all the more so because card number four (maximum I. H. P.) was taken when wind and sea were both moderate, while card number twelve (maximum speed) was taken with a rough sea and wind of force seven.

Mr. Dickie has misunderstood our remark about the error in taking the counter. It was assumed that there might be an error of one revolution in reading the counter, and this error would be just the same, that is one revolution, for four hours as for the three minutes or so on the mile. Of course, if an error is made on the mile, it would be multiplied in computing the number of miles. His remark in regard to the possible error in laying out the revolution curves is very important, and it is for this, among other reasons, that we would insist on the importance of chronographic records for the measured mile runs. Except for the purposes of record, it would not be necessary to go to this

trouble on the trial, as ordinary stop watches and duplicate counters would be sufficiently accurate. Mr. Mattice's suggestion of taking flash light photographs of the counter is excellent, and could be very easily arranged.

With regard to the fourth method (the run over a straightaway course), it has been apparently overlooked by those favoring it that it is not a very easy matter to lay out such a course without an error of sufficient magnitude to render the result unsatisfactory. I have been informed that the Coast Survey will not guarantee a ten-knot course with a smaller probable error than 150 feet. This would amount to about five-hundredths of a knot per hour for a twenty-knot ship. This difficulty would be obviated, however, if the contractors should agree to accept a carefully laid out course as correct. One disadvantage of this method, which the third completely avoids, is that it does not allow of extending the trial for several hours if it is found that the ship is continually improving. The importance of this will be realized when it is remembered that on the *Charleston's* trial the last four out of six hours were chosen. The only way, in the fourth method, would be to turn and run back over the course, and, as its continuity would be broken by the turn, it would cease to be a straightaway trial. In the third method, the trial can be kept up as long as the contractors desire and circumstances will permit.

It would be a mistake, in my judgment, to increase the base for the preliminary runs to more than a knot. As soon as this is done, current measurements must be added to the other data to be taken with great accuracy, as is pointed out by Mr. Dickie; and the difficulty will be introduced of finding such a base in deep, sheltered water.

Whatever method the Department may adopt, the record of the trial, combined with data from progressive trials afterward, will enable a very good idea of the value of the third method to be formed, and when the *Philadelphia* and *San Francisco* shall have been tried, I shall be glad to submit a note for the JOURNAL on this subject.

[The preceding paper was sent to Chief Engineer Isherwood for discussion and comment, and, as the method he gives had not been touched upon, and some other items only noticed casually, he very kindly sent the following article as an expression of his views and a discussion of the whole problem. It was not received in time to be sent to those gentlemen who have discussed the first paper.—EDITOR.]

IV.

HOW TO ACCURATELY MEASURE THE SPEEDS OF LARGE, FAST VESSELS DURING SEVERAL CONSECUTIVE HOURS.

By Chief Engineer Isherwood, U. S. Navy.

Among the most important data required to be known in the present state of steam engineering science, is the *exact* speed of steamers, in the case of which a mere reasonable approximation to the truth cannot be accepted, because, as the speed enters into the economic and potential performance of the machinery in the triplicate ratio, a small error in the direct determination involves an exaggeration of that error in respect of this performance in the ratio of its cube. For example, an error so small as 2½ per centum in the speed of the vessel, an error so easily made with the best precautions that there is grave doubt whether the speed of any steamer is known with certainty within less than that limit, becomes an error of ($1.025^3 =$) 7⅔ per centum as regards the effect of the power applied to the propulsion of the vessel.

The experienced experimenter knows that there is nothing so difficult as the making of *exact* physical measurements in any science, even with the best means and appliances directed by the best trained judgment, particularly when the thing measured is large and the measuring instrument small, or when the measuring has to be done indirectly, or by inference from other than direct observation, or when variable causes are concerned.

Now, in the case of a steamer, the speed required is the speed through the water uninfluenced by wind or wave, and as there is no water navigated by steamers that is not so influenced, the ascertainment of this speed becomes not only difficult, but proportionally difficult as these influences are greater and more variable. Obviously, in ascertaining the speed of a steamer, a place for the trial should be selected where there is neither cur-

rent, wave nor wind, but, as such a place can scarcely ever be found in practice, the experimenter is compelled to make use of whatever locality can be commanded, and to eliminate the influences of the winds, waves and currents he may find there by the best methods his knowledge can devise. Large steamers especially are unwieldy, inconvenient and very expensive to experiment with, requiring much time to determine a few facts. Their draught of water is considerable, and as both the depth of water beneath their keels, and the breadth of the water in which they are tried, affect, when too small, the result enormously and prejudicially to the performance of the vessel, the locality must have such a depth and breadth of water as will sensibly eliminate these causes of error. This is a *sine qua non.* For a large vessel at high speed, there should not be less than twice the vessel's draught of water beneath her keel, and the breadth of the water should not be less than two-thirds of a mile. As the size of the vessel and its speed decrease, these proportions can be decreased. In all cases the surface of the water should be smooth, and this condition is of the greater importance the smaller the vessel. The wind should not exceed a light breeze, and its direction relatively to the vessel should be such as not to produce any sensible careening of the hull. The effect of the breeze can be nearly eliminated by running the vessel alternately with and against it, and the same can be said with regard to tide or current when their velocity is uniform, taking care to make the calculations *for equal time* run in opposite directions, and *not for equal distances* over the ground. In all locations practically available for experimenting on the large scale, there are always variable wind and current.

For the accurate determination of speed, a base of known length on the shore is absolutely essential. If the tide or current were uniform, and also the wind, the longer the base the more accurate would be the result, because the errors of "ranging and timing" occur at the ends of the base. These errors are quantitatively the same, let the speed be what it may, and, consequently, proportionally greater the greater the speed for a given base, because less time then elapses during a run over the

base. The "ranging" error being positive, is the greater proportionally the shorter the base. But when the current is variable, as in the case of tide, which is so far from being uniform that it is continuously changing from nothing to about three miles per hour, the longer the base the greater the error from this cause, for it affects the apparent speed unequally for equal time on the alternate runs over the base. Hence, a compromise must be made on length of base. For vessels of medium speed, a base of two miles is sufficient, while for fast vessels three miles are not too much. The ranging poles at each end of the base must range *exactly* at right angles to the base, or serious error will result, particularly when the vessel is tried at different distances from the base.

As regards the proper number of double runs that should be made over the base to furnish a correct mean, the greater the number, without limitation, the more accurate will be the mean. For extreme accuracy, not less than ten double runs should be made for each intended speed, the revolutions of the screw or paddle-wheel per minute being kept as constant as possible. Any variation in this latter particular tends to falsify the result whenever the vessel's speed is great enough to cause her to "squat" at the stern, as after that phenomenon occurs the resistance of the vessel increases in a higher ratio than the duplicate ratio of the speed. Particular care must be taken that the vessel pass *both* ends of the base at the intended speed, so that there will be no error due to increase or decrease of speed at those points.

The speed of the vessel determined by the mean of, say, ten double runs over the base with all the power that could be forced from the machinery, is *the speed for the running time only*, and and there cannot be inferred from the result obtained in such manner that this speed could be sustained for any much greater length of time. In so short a trial, the best quality of coal, handpicked, consumed at the maximum rate of combustion by artificial draught in clean furnaces, the heat absorbing surfaces of the boilers being also quite clean, would be used by an unusually large number of firemen or stokers with unimpaired

strength, and their efforts stimulated by promise of reward. The lubrication of the working parts of the engine would be copious, the adjustment of those parts would be perfect, and the supervision of the machinery would be made by superior skill embarked for the special occasion, so that the speed thus obtained must be considered not the *maximum*, but the *maximum maximorum*. The true maximum speed of a steamer is the speed which, uninfluenced by wind or wave, can be continuously maintained by the engineering staff permanently embarked, with commercial coal, and as long as it lasts.

To properly ascertain the performance of a screw steamer, the pitch and other dimensions of the screw must be exactly measured on the casting, and for each blade separately. The counter must be reliable, and the indicators tested and their coefficients ascertained. These things are rarely done. Moreover, and what is quite as important, absolute authority should be given the experimenters until the experiments are completed, so that they can be held responsible for either errors or lacunes, as well as to be able to accomplish the object satisfactorily. Above all, the experimenters must be experienced and competent. An experiment made in a perfunctory manner by incompetent persons will be simply misleading. The *personnel* are quite as important as the *materiel*.

The essentials, then, of a proper location for a base are:

1st. Sufficient length in a straight line.
2d. Sufficient depth and breadth of water.
3d. Smooth water.
4th. Absence of wind.
5th. Minimum tide or current.

To find a location proper for experiments with large, fast steamers, is very difficult, and the difficulty becomes well nigh insuperable when the trial is to be continuous for several hours in order to fulfil a contract stipulation. A satisfactory location for a base, however, can be found with comparative ease in land-locked positions for small vessels of moderate speed, and a trial limited by length of base.

Now the object to be reached is, whether determinations from

such a base by means of such a vessel cannot be accurately applied to determining the speeds of large, fast steamers in a smooth sea, with winds not exceeding light breezes, and irrespective of any base there, so that if the trial was to be of four or six continuous hours it could be made in two consecutive straight runs in exactly opposite directions, each run consuming half the contract time, exclusive of the time in the one turning not counted. This can certainly be effected, and in the following manner:

The best practical method of ascertaining the velocity of a fluid, liquid or aeriform, or, conversely, of ascertaining the velocity of a solid through a fluid, when the velocities cannot be measured by direct observation of time elapsed while passing between fixed points a known distance apart, is by means of an immersed screw having its axis placed horizontal in the fluid in the direction of the movement. The number of revolutions made by such a screw while the fluid or solid moved a given distance, as measured by stationary marks a known distance apart in some location which permitted such observations to be made, gives the distance moved by the screw, as a mass, in the fluid per revolution in units of length. The distance thus advanced by the immersed screw per revolution being known, the application of this system of measurement for measuring the velocity of fluids, or of solids moving in fluids, in cases where direct observation is impossible, becomes very easy. All that is necessary is to count the number of revolutions made by the screw per unit of time by the distance passed per revolution, and the product is the velocity required. The distance advanced by the screw per revolution under these circumstances is always more than its pitch, generally from one-fourth to one-third more, according to the ratio of its pitch to its diameter, because the molecules of water pressed by the forward side of the blades, as the screw is dragged through the water, move forward, owing to their mobility, and the screw, as a consequence, advances per revolution not only the distance (its pitch) it would were there no such forward slip, but additionally the forward recession of the pressed molecules of water or distance of that slip per revolution. This action is the reverse of what takes place with a propelling screw,

but is due to the same cause, namely, the mobility of the water. In the case of the propelling screw there is an aft slip, so that the vessel or screw never advances a distance per revolution of the screw equal to the pitch of the screw, while, in the case of the measuring screw, there is, on the contrary, a forward slip, so that the screw always advances a distance per revolution greater than its pitch.

The measuring screw, after being rated, when employed to denote the speed of a vessel, must not be placed either forward or aft, but on the side of the vessel at about the center of its length and not less than six inches in the clear from the side. It should be placed several feet below the surface of the water, so as not to be affected by the surface currents due to the movement of the vessel.

The bow of the vessel raises the water displaced by the vessel's movement above the general water level, after which this raised water flows out over that level in directions at right angles to the water lines of the bow, causing a surface current. At the stern of the vessel, the water replacing the water displaced at the bow, ascends vertically from the level of the upper line of the keel, and rises above the general water level by its momentum, spreading out in all directions over that level and thus causing a surface current at the stern. This action is intensified in the case of a screw vessel, because the water displaced by the slip of the screw also ascends vertically after it is clear of the constraints of the screw and vessel, and rises by its momentum above the general water level. Below the water level, however, there is scarcely any disturbance due to horizontal currents. Such currents exist only at the surface of the water and are due to local elevations of the water above the general level. No inferences are warranted of what obtains below the surface by what appears above the surface.

For the reasons just stated, the indications of no log, whether of the common "chip" kind, or of what is termed the "patent" log or "taffrail" log, are reliable within considerable limits. These logs are surface logs, and they float in the surface current which exists at the stern of all advancing vessels owing to the

causes described. If they were exactly rated for smooth, quiescent water, they would give too large indications when placed in the surface current flowing aft from a vessel, and the error would be as variable as the current. In the case of the "chip" log, almost universally employed during centuries for ascertaining the speed of sea-going vessels, the experienced officer using it always deducts a certain quantity from its indications, according to his judgment, based on the speed of the vessel and other variable circumstances. In a word, he throws the log and from it guesses at the speed. The "patent" or "taffrail" log, which is a measuring screw dragged at the surface of the water abaft the vessel and in the aft flowing current there, transferring its revolutions through the cord which drags it, to an index on the taffrail of the vessel showing the speed, is, for the same causes, unreliable and variable in its indications. Both kinds of log would give more accurate indications were they placed in the water on either side of the wake of the vessel, but to do this would require a very troublesome and inconvenient arrangement of projecting platforms, etc. It should be done, however, in all cases where practicable, and, particularly, in trials of vessels in smooth water to determine speed in fulfillment of contract stipulations. In the latter case, if the ordinary logs are employed, the chip log should be continually thrown from a platform projecting from one side of the vessel, and the taffrail log should be dragged from a similar platform on the other side, the indications of the one checking those of the other. In many of the late trials of four and six hours' duration, made with recently constructed large and very fast screw steamers by European governments, the speeds as reported being obtained from taffrail logs, are probably from one to two miles per hour too great, an error very easy to make, very gladly accepted, and never corrected or explained. In this way there sometimes results a preposterous negative slip by the screw, and an equally preposterously small resistance by the vessel when its alleged speed is compared with the alleged power developed by its engines. Some, and particularly one, of the various modifications of Pitot's tube, have proved successful as accurate measures of low speeds under exceptional conditions

of smooth water and no wind or current, but never for high speeds. None of them could be applied with confidence to determine the speed of a fast steamer at sea under the adverse conditions of current drift, heeling by the wind, and rising and falling with the swell, however gentle. The writer has used the best modification of this tube, and can speak of its limitations from experience.

The only reliable and accurate method of measuring the speed of a vessel when a satisfactory base cannot be obtained, is by a strongly made screw of sufficient dimensions to permanently retain its form, and the position of its axis both horizontal and parallel to the vessel's keel, and sufficiently submerged to prevent its exposure by any rolling or heeling of the vessel, or by any undulations of the waves. The revolutions of such a screw are to act by means of rigid connections giving positive motions upon a counter secured upon the vessel's deck, and the number in a given time of these revolutions so denoted, allowing for the effect of the gearing, and multiplied by the screw's advance through the water per revolution, ascertained as hereinbefore described, will give the vessel's speed.

The screw is to be placed about midlength and on one side of the vessel, and should be made of bronze and as light as sufficient strength will permit. It should be a true screw, about 16 inches in diameter, 6 feet in pitch, 4-bladed, 6 inches in length, and of uniform length from the hub to the periphery. The hub should be about 2 inches in diameter and perforated by a hole 1¼ inches in diameter. An extension of the hub at both ends allows journals of 2½ inches length to be made with ample collars to hold against the pressure of the screw during the going ahead or backing of the vessel. The journals are to revolve in lignum vitæ bearings, and the collars are to revolve against the same material. All the fittings of the journals and collars are to be quite loose, so as to avoid friction as much as possible. One end of the hub is to be extended far enough beyond its journal to receive a vertical miter toothed wheel gearing into a horizontal miter toothed wheel of twice its diameter at the pitch line. From the center of the horizontal wheel a hollow brass shaft

(tube) of about 2½ inches diameter extends vertically to the deck of the vessel, and the top of this shaft by means of a crank upon it, or other device, gives the necessary reciprocating motion to an ordinary engine room counter secured on the vessel's deck. The bearers of the journals, the thrust collars, and the supports for the toothed wheels, are to be held by a metal frame of sufficient stiffness lying against and supported by the side of the vessel, and extending from the screw to the deck. The frame on both sides at the bottom is to slide into a short groove made in a strip of metal bolted to the side of the vessel. This frame can be easily lifted out of the water bodily with its screw to the deck whenever an examination of the screw is deemed necessary. If preferred, a worm and wheel can be substituted for the two miter gear wheels; and a continuous counter formed by a train of geared spur wheels can be substituted for the reciprocating engine room counter.

The screw should, in a large vessel, be placed about six feet beneath the water, to secure its being always submerged, but the depth does not affect its action or the constancy of its indications. The gearing forms an easy method of changing the vertical revolutions of the screw shaft into the horizontal revolutions of the shaft of the large miter wheel, and the larger diameter of this wheel reduces the number of strokes to be made per minute by the counter within a safe practical limit for even the highest speeds of vessel.

When the above described apparatus is employed, the vessel being in a smooth, open sea, the course should be chosen at right angles to the direction of whatever light breezes there might be, so as to neutralize their effect. If such a course should not be convenient, then the vessel should be run half the time with and the other half against the wind. And if the wind exceed a light breeze, this alternate running should be made, in order to avoid the injurious effect upon the speed of the heeling over or careening of the vessel by the wind.

The accuracy of the measuring screw, or the constancy of the number of its revolutions per unit of length passed by it through the water, is not only not affected by the depth of its submerg-

ence, but is not affected by its speed through the water after the speed has been acquired which causes it to revolve uniformly. A certain speed is required—from about 2½ to 3 miles per hour—to cause the screw to begin revolving, but after the revolutions become regular, at say 4 miles per hour, they remain sensibly unchanged in number per unit of speed through the water at all higher speeds. In other words, twice the number of revolutions of the screw per minute indicate twice the speed of vessel, and so on in direct ratio.

Except for the frictional resistances developed by the movement of the screw, it would advance bodily through the water with the speed of its form, namely, the product of the number of revolutions made by it in a given time into its pitch; but the effect of these resistances is to retard the revolutions due to the reaction of the water were there no frictions, so that the speed with which the revolving dragged screw bodily passes through the water is greater than the speed due to its form, and as a consequence a mass of water is carried before, or pushed forward by, the screw, which mass offers a resistance proportional to the projected area of the screw on a plane at right angles to its axis, multiplied by the square of the velocity of the drag of the screw.

The drag of the screw when revolving freely by the reaction of the water against the front of its blades is the difference between the distance advanced by its form in a given time and the distance through which it advances bodily in the same time, the latter being the distance advanced by the vessel dragging it. The drag is expressed in fractions of the latter distance.

The form of the screw, that is to say, the obliquity of its helicoidal surface as determined by the ratio of its pitch to its diameter, has no effect upon the drag of the screw, which consequently remains sensibly constant for all screws; the number of revolutions made in a given time by dragged screws of different pitches being in the inverse ratio of the pitches; hence, the drag of the screw is, in all cases, the same per centum of the speed of the vessel dragging it.

The results of numerous experiments with dragged screws

of enormously differing dimensions, give the drag at one-third of the speed of the vessel dragging them; but, as this proportion is derived wholly from experiment, and may admit in particular cases of slight variation, the number of revolutions made by the measuring screw per mile through the water should always be ascertained by experiment over a carefully determined base.

The forces resisting the revolution of the screw when it is dragged through the water, are:

1st. The friction due to its weight upon its journals. This friction is constant at all speeds of revolution.

2d. The resistance of the counter to movement. This resistance is constant at all speeds of revolution.

3d. The friction of the toothed gearing under the strain required to work the counter. This friction is constant at all speeds of revolution.

4th. The friction due to the weight of the gearing and the weight of its connections. This friction is constant at all speeds of revolution.

5th. The friction due to the direct resistance against the water of the vertical revolving shaft of the horizontal miter wheel. This friction increases as the square of the speed of the vessel, because the water pressure producing it increases in that ratio.

6th. The friction due to the direct resistance of the vertical revolving miter wheel against the water. This friction increases as the square of the speed of the vessel, because the water pressure producing it increases in that ratio.

7th. The friction due to the resistance to the water of the wetted helicoidal surface of the screw. This friction increases as the square of the speed of the vessel.

8th. The friction due to the endwise pressure of the screw against the collars of its shaft caused by the drag of the screw. This friction increases as the square of the speed of the vessel.

The difference between the speed of the form of the dragged screw, and the speed of the vessel dragging it is caused entirely by the above frictions. Could they be eliminated, the revolutions of the dragged screw would not be in the least retarded, but would be what was due to the quotient of the speed of the

vessel divided by the pitch of the screw. The retardation is only effected by opposing forces acting in the direction of the revolution of the screw. The resistance of the wetted helicoidal surface of the screw acting on both sides of the blades and in the directions of the helices of the screws, does not affect the revolutions of the screw by the reaction of the water. Of course, the resistance of the wetted surface requires power to overcome it, and this power is taken from the moving vessel, whose speed is consequently lessened by a quantity equivalent to this resistance, but the revolutions of the screw are not affected. The blades of the screw being gradually tapering from the center to the edges, front and back, where the thickness of the metal is reduced to a line, there would be no resistance by the front edge of the blades, and for the same reason there would be no resistance caused by the displacement of water by the bulk of the blades, as they are wholly submerged, consequently there would be no power expended on account of these, and the vessel would not suffer any diminution of speed.

The force causing the screw to revolve is derived from the pressure of the water on the front of the blade, produced by the movement of the vessel. This pressure is in the ratio of the square of the velocity of the vessel, but the portion of it applicable to revolving the screw is the difference between it, and the aggregate pressure produced in the opposite direction by the retarding forces just described. Now if these latter forces were also in the ratio of the square of the velocity of the vessel, then both the force producing the revolutions of the screw, and the forces retarding these revolutions, would remain in the same ratio at all speeds of vessel, and, consequently, the number of revolutions made by the screw per unit of time would be in the direct ratio of the speed of the vessel; or, the number of revolutions made per mile of the vessel's movement through the water would be constant. But if, on the contrary, the retarding forces increased in a less ratio than the accelerating force, then the number of revolutions made by the screw per unit of time would increase with each increase in the vessel's speed, or, the number of revolutions made per mile of the vessel's movement through

the water would increase with increase of speed, and the drag of the screw would proportionally lessen, because the portion of the water pressure produced by the movement of the vessel, applicable to revolving the screw, would then increase with increased speed of vessel.

Now, of the foregoing eight enumerated forces resisting the revolution of the screw, the 1st, 2d, 3d and 4th, are in the simple ratio of the speed of the vessel; and the 5th, 6th, 7th and 8th, are in the duplicate ratio of the speed of the vessel. But the first four are too insignificant as quantities in comparison with the last four to much modify the latter, so that practically the retarding forces may be taken as being very closely in the same ratio as the accelerating force, especially as this last does not increase in quite the ratio of the speed of the vessel. It would do so did the water pressure on the back of the blades remain constant at all speeds of vessel, but that pressure decreases as the speed of the vessel increases in some unknown proportion, owing to the inability of the water to fill in "solidly" against the back of the blade as the screw is dragged along. Hence, there results that the accelerating force increases in the ratio of the square of the speed of the vessel plus a comparatively small force increasing in some less ratio than that of the square, which just about equalizes the slightly less ratio than that of the square with which the resisting forces increase with increased speed of vessel, so that the equality between the ratios with which the retarding and accelerating forces increase with increased speed of vessel remains unimpaired.

Hence, when the measuring screw is dragged through the water at what speed it may, above the low speed required to begin its turning, it will make the same number of revolutions per mile; or, the number of revolutions made by it per unit of time will be in the ratio of the speed of the vessel; or, the drag of the screw expressed in lineal units will be in the ratio of the speed of the vessel; or, the drag of the screw expressed in fractions of the vessel's speed will be a constant for all speeds. These results have been practically verified whenever such screws have been tested by the writer at widely different speeds. He

has experimented with them extensively, and can recommend them as the most convenient, the cheapest and the most reliable means for obtaining the exact speeds of vessel when a suitable base cannot be employed.

The propelling screws of seagoing vessels, uncoupled and revolving freely by the reaction of the water when the vessels are under sail alone, are measuring screws on a large scale, and the number of revolutions made by them per unit of time has been found to be in constant proportion with the speed of the vessel at all speeds, and in harmony with the principles just shown. These screws being submerged, are not affected by the horizontal surface currents of water at the sterns of the vessels. Formerly all naval screw vessels were fitted with full sail power and disconnecting clutches for their screws, which latter were allowed to revolve freely in order to diminish their dragging resistance when the sails were used without them, which was for a large portion of the time. An investigation of the ratio between the revolutions of the screws and the simultaneous speeds of the vessels, from the logs of a considerable number of these vessels, fully confirms the laws herein stated, and the constancy of the ratio at all speeds. None of these uncoupled screws commenced revolving until the vessel had a speed of about three miles per hour. At this speed the reacting force of the water on the front of the blades was just equal to the force of the various frictions enumerated, and, after revolution commenced, these frictions were sufficient to retard the number of revolutions which would otherwise be made by the screw, one-third, that is to say, the distance advanced bodily by the screw per revolution when dragged by the vessel was one-half more than the pitch of the screw. When the pitch, therefore, is known of either the measuring screw or of the propelling screw when dragged, and the number of revolutions made by them during, say, one hour, the speed of the vessel dragging them will be obtained by multiplying the pitch by the revolutions and adding one-half the product.

Although obtaining a suitable base for the speed trial of a large, fast vessel during several consecutive hours may not be

practicable, the obtaining of one suitable for a speed trial of a small or medium sized vessel at moderate speed for the purpose of "rating" or determining the drag of a measuring screw presents no difficulty anywhere. A proper base of one, or preferably two, miles in length, straight, with smooth water and not much variation of tide, and having sufficient depth and breadth of water for the vessel's draught and speed, being selected, and a time when the wind does not exceed a light breeze being chosen for the trial, the vessel, having the measuring screw to be rated properly affixed to its side, is to be run over the base about twenty times alternately in each direction, at the speed of about five miles per hour, the time and the number of revolutions of the measuring screw during each single run being accurately ascertained. Then these runs are to be repeated at about eight miles per hour speed of vessel, and the same determinations made. Finally, these runs should be again repeated with the speed of vessel of about eleven miles per hour, and the same determinations made as before. The three different speeds will show whether the revolutions of the measuring screw per mile passed through the water by the vessel are affected by the speed of the vessel. If the experiments be accurately conducted, only very small differences in the number of revolutions made by the measuring screw per *double* run will be found, be the speed of the vessel what it may, and these differences will be found quite irregular in their distribution, and without regard to the speed of the vessel, but due to accidental causes.

A very simple and decisive test of the reliability and accuracy of the measuring screw can be made by securing firmly the helm of the vessel at an angle of, say, 30 to 35 degrees with the keel, and then making a number of circles with the vessel in succession, fixing the beginning and end of each circle by the indications of a sensitive compass, when the measuring screw will be found to make exactly the same number of revolutions per circle, and this number will not be affected by either wind, wave or current. The measuring screw is, indeed, the only practicable method of determining the diameters of circles made by vessels

turning by means of their rudders placed at different angles with their keels.

When the experiment over a base is made with a variable current or tide, and supposing the engine to work with a constant number of revolutions per minute, the time of making each run will vary as the current varies; hence, the proper method of determining the correct speed of the vessel from the apparent speed becomes important.

In this connection there must be kept in mind that each two successive runs over the base will be made with and against the current, and the correct speed deduced from a double run must be assumed to be the arithmetical mean between the two apparent speeds. Now, as in the case of any two consecutive runs, one run will be with and the other against the current, a given number of double runs allows a greater number of determinations than the number of double runs. For example: two double runs give three determinations of speed; three double runs give five determinations of speed; four double runs give seven determinations of speed; five double runs give nine determinations of speed; six double runs give eleven determinations of speed, and seven double runs give thirteen determinations of speed, etc., so that doubling the number of double runs gives one more speed determination than twice that number of runs. For each run the determination must be made of the apparent speed of the vessel *per hour*, and then the mean of these apparent speeds for each two consecutive runs must be taken for the true speed during that particular double run. After these true speeds have been ascertained for each double run, the arithmetical mean of all of them will be the true speed of the vessel during the experiment.

The following table will illustrate the correct method of obtaining the true speed:

Designation of the runs.	Time in minutes and seconds of making the runs over a base of one mile.	Apparent speed of vessel in miles per hour for each run over the base.	True speed of vessel in miles per hour for each double run over the base.
1	5:00	12.0000	
			13.5000
2	4:00	15.0000	
			13.6225
3	4:54	12.2450	
			13.2653
4	4:12	14.2857	
			13.3498
5	4:50	12.4138	
			13.4358
6	4:9	14.4579	
			13.5447
7	4:45	12:6316	
			13.4586
8	4:12	14.2857	
			7)94.1767
True mean speed in miles per hour, ...			13.4538

The true number of revolutions made by the measuring screw per unit of time, corresponding to the immediately above true speed of the vessel, will be obtained in a similar manner, being determined first in the case of each run for the number of revolutions made per hour, and then for the mean number of revolutions made by the measuring screw per hour per double run of the vessel over the base, and, finally, taking the arithmetical mean of all the latter determinations for the true number of revolutions made by the measuring screw per hour during the experiment, as illustrated in the following table:

Designation of the runs.	Time in minutes and seconds of making the runs over a base of one mile.	Total number of revolutions of the measuring screw while passing over base.	Number of revolutions of the measuring screw per hour for each run over base.	Number of revolutions of the measuring screw per hour for each double run over the base.
1	5:00	750	9,000	
2	4:00	600	9,000	9,000
3	4:54	735	9,000	9,000
4	4:12	630	9,000	9,000
5	4:50	725	9,000	9,000
6	4:9	622	9,000	9,000
7	4:45	713	9,000	9,000
8	4:12	630	9,000	
				7)63,000
True number of revolutions of the measuring screw per hour, ...				9,000

Dividing the above 9,000 by the 13.4538 miles per hour speed of vessel, there results 668.956 revolutions of the measuring screw for each mile passed through the water by the vessel. This may be taken as the "rating" of the measuring screw, and, when obtained, by experiment with a small or medium-sized vessel at medium speeds over a base, the screw can be attached to the side of any large, fast vessel for speed trial in localities where a suitable base cannot be had, and it will measure correctly the speed of such vessel.

One of the objections to measuring the speed of a large, fast vessel directly over a short base, even if the locality affords sufficient depth of water, is that the breadth of water will not be sufficient, for the vessel must be run near enough to the ranging poles which mark the ends of the base, to enable the observer

to distinctly see them and range them without too much error. The farther from the poles, the greater is the "ranging" error; and the nearer to the poles the less is the breadth of the water between ship and shore. Also, the shorter the base and the faster the vessel, the greater is the "timing" error. These objections can be obviated only by employing a measuring screw and "rating" it by means of a small or medium-sized vessel, tried over a short base at not above medium speeds. With such a screw, the large, fast vessel can be tried in any smooth, open sea for any length of time, and the results confidently depended on.

As an illustration of the errors probable to be made in the determination of the speeds of even medium sized vessels with medium speeds, the case may be given of the trials of the United States screw steamer *Boston* made on the 3d of August, 1889, over the measured mile at Newport, R. I., with what is considered (not by the writer, however,) satisfactory precautions for accuracy.

The trials were at four different speeds, four runs being made over the base for each speed, and the means taken for that speed.

The water was smooth and very deep (19 fathoms minimum depth), the course straight, and the bottom of the vessel perfectly clean. The screw has a fore-and-aft expanding pitch—forward pitch 22.819 feet, aft pitch 25.865 feet, by measurement, mean pitch 24.342 feet. During the maximum speed the air pressure in the closed fire room was equivalent to that of a column of water 1⅔ inches high.

The results are given in the following table, and particular attention is called to the considerable discrepancies in the slip of the screw for the different speeds of the vessel. Now, as the maximum speed of the vessel was within the "squatting" limit, the slip would have been the same for all four speeds, had they been truly determined; the differences of the reported slips mark, therefore, the variations from the truth. As the mean from a greater number of trials is, *ceteris paribus*, proportionally more reliable than the mean from a smaller number, the proper method, in examining this case, is to take the mean of all the

slips as the true slip, namely 11.75 per centum of the axial speed of the screw calculated for the after pitch, all the slips having been calculated for that pitch, and then redetermining the speeds of the vessel for this constant slip. The differences between the speeds so determined and those reported experimentally will show the experimental errors.

Number of revolutions made by the screw per minute.	Experimental speed of vessel per hour in geographical miles of 6,086 feet.	Experimental slip of the screw in per centum of its pitch.
72.2	15.585	15.375
64.8	14.350	13.183
51.2	12.000	8.115
21.1	4.610	14.346

Calculated speed of vessel for constant slip of screw of 11-.75 per centum of its speed in geographical miles of 6,086 feet per hour.	Difference between experimental and calculated speeds of vessel in fractions of a mile per hour.	Difference between experimental and calculated speeds of vessel in per centum of the former.
16.253	0.668	4.286
14.587	0.237	1.652
11.525	0.475	3.958
4.750	0.140	3.037

Judging from the above, the probable error for a vessel of 19 miles per hour true speed would be over six-tenths of a mile, a very serious quantity when $50,000 are to be paid for every quarter of a mile. The difference due to such error would be $120,000 either in favor of or against the Government, and this error may easily be exceeded in the case of a large, fast vessel when tried in the manner the *Boston* was, and which is, indeed, the usual manner, and made as carefully, no doubt, as such trials are usually, and by the persons who conduct them.

V.

TURKISH TORPEDO BOATS.

TRANSLATED BY ASSISTANT ENGINEER EMIL THEISS, U. S. NAVY.

[The boats herein described were built by the Germania Ship and Engine Building Co., Berlin and Kiel, and the description of their hull and machinery is taken from *Die Zeitschrift des Vereins Deutscher Ingenieure.*]

The program for the increase of the Turkish navy, adopted some years ago, called principally for fast cruisers and sea-going torpedo boats similar to those already possessed by a number of naval powers. At that time three torpedo boats were contracted for with a French establishment and five with F. Schichau. of Elbing. The German boats seem to have proved the more satisfactory, for soon after (in 1889), orders for nine more torpedo boats and three cruisers of different dimensions were placed with the Germania Company. Five torpedo boats have been completed. Two of these, after having met all contract requirements on their trial trips, proceeded by sea to Constantinople, meeting with pretty rough weather on the way and proving themselves good sea boats. The other three are at present undergoing their trials.

Hull.—Length over all, 128 feet; beam (greatest) 15′ 9″. Completely equipped for service they draw 3′ 5″, displacing about 85.6 tons. The hull is built entirely of mild steel of German manufacture. The following are some of the dimensions and scantlings: The frames, which are angle irons, $1\frac{3}{16}'' \times 1\frac{5}{16}'' \times \frac{5}{32}''$, are spaced $19\frac{11}{16}''$ apart. Under the engines they are formed of two such angles placed back to back. The reverse frames are angle irons $1\frac{3}{16}'' \times 1\frac{3}{16}'' \times \frac{5}{32}''$. They are doubled in the engine and boiler space and their dimensions increased to $1\frac{37}{64}'' \times 1\frac{37}{64}'' \times \frac{5}{32}''$. The transoms are $7\frac{7}{8}''$ deep; in the engine and boiler space they are $\frac{5}{32}''$ thick; forward and aft they are

$\frac{1}{8}''$. The deck beams are angle irons, $1\frac{3}{16}'' \times 1\frac{3}{16}'' \times \frac{5}{32}''$ amidships and $1\frac{3}{16}'' \times 1\frac{3}{16}'' \times \frac{5}{32}''$ at the ends. The hold stringers, likewise angle irons, are $1\frac{3}{4}'' \times 1\frac{3}{4}'' \times \frac{3}{64}''$ amidships and $1\frac{37}{64}'' \times 1\frac{37}{64}'' \times \frac{5}{32}''$ at the ends. The keel plate is $\frac{11}{64}''$ thick amidships and $\frac{5}{32}''$ at the ends. The doubling plate is $7\frac{7}{8}''$ wide and $\frac{13}{64}''$ thick. The sheer strakes decrease from $\frac{13}{64}''$ amidships to $\frac{5}{32}''$ at the ends. The rest of the outer plating has a general thickness of $\frac{1}{8}''$. The deck plates outboard above the engine and boiler compartments are $\frac{5}{32}''$ thick. All the rest is of a thickness of $\frac{1}{8}''$. The watertight bulkheads are built of steel plates $\frac{1}{8}''$ thick below and $\frac{5}{64}''$ above, stiffened by $1'' \times 1'' \times \frac{1}{8}''$ angles.

Subdivisions of Hull.—The boats are divided by seven watertight bulkheads into eight watertight compartments. The first compartment from aft is accessible from the store room forward of it through a manhole with a screw-down cover. This compartment is usually empty and serves no specific purpose.

The second compartment is the store room. It is fitted with shelves and lockers, and is accessible from the upper deck through a watertight hatch. This compartment also contains the magazine for the after rapid-firing gun.

The third compartment contains the quarters, described further on, for the commander and the machinists.

The engine room forms the fourth compartment. It is reached through a hatch closed watertight. A skylight above the engine furnishes light and accommodates the engine parts projecting above the upper deck.

The fifth and largest compartment contains the boiler and coal bunkers. It is reached from deck through a hatch with watertight cover, and from the engine room through an air-lock. A skylight furnishes light. The coal bunkers, one at each side of the boiler, have, including the reserve bunkers, a capacity of about twenty tons.

The sixth compartment is the torpedo room, and contains the crew's quarters. At the after end on the port side is the galley, on the starboard side the water closet. Between the two, below the floor, is stowed the ammunition for the forward rapid-firing

gun. The air-compressing pump and the air reservoir are also located in this compartment.

The seventh compartment, accessible from deck, through a manhole, contains the cartridge for discharging the torpedoes, to be described further on, and the bow rudder.

The eighth or bow compartment is empty, and is reached from No. 7 through a manhole, closed watertight.

Accommodations for officers and crew.—From the after conning tower a ladder leads into the officers' quarters in compartment No. 3. The wardroom is comparatively spacious, well lighted and ventilated by a skylight overhead, and nicely finished and furnished. Forward of the ward-room are two state rooms, separated from the former by a water closet on the port and a pantry on the starboard side.

The crew's quarters, in compartment No. 6, are reached through the forward conning tower. They are provided with ample locker room and the usual hammock hooks. The crew numbers 14 men, not counting the commander and the machinist: two petty officers to assist the commander, two to assist the machinist, four firemen, and six deckhands.

Boiler.—Steam is furnished for the main and auxiliary engines by the locomotive boiler shown on Plate II. Working pressure 185 pounds. Grate surface 35.5 square feet. Heating surface 1,560.78 square feet. As the engines developed over 1,300 I. H. P. on trial, 36.6 H. P. per square foot of grate was obtained.

Main Engine.—The engine is vertical, triple expansion, of about 1,300 I. H. P. The cylinder diameters are 16.″14, 24.″41 and 35.″43, with a stroke of 16.″53. As is usual on torpedo boats, the cylinders are unjacketed. The angle between the cranks is 120 degrees, the H. P. crank leading. Marshall's valve gear is used. Its use offers a wide range for expansive working, the power having been made to vary in this way from 213 to 1,213 I. H. P. The condenser is made large enough to condense all the steam that the boiler can make, and that is capable of direct discharge into the condenser.

Pumping Engine.—In order to make the main engine as

simple as possible, it is, unlike most other torpedo boat engines, including those of the German navy, used for the single purpose of driving the propeller. The pumping engine is triple expansion; the H. P. and I. P. cylinders, placed one above the other, act upon one crank, and the L. P. cylinder upon another at right angles with the former. This engine drives the air pump, a centrifugal circulating pump, two vertical double acting feed pumps, and two similar smaller pumps. Of these, one is the engine room bilge pump, the other a pump for circulating cold water through the cap brasses of the crank shaft bearings and through the crosshead guides.

The following advantages result from the use of a separate pumping engine:

1. Moderate piston speeds at maximum power. In the present case the number of revolutions of the pumping engine did not exceed 150, corresponding with a piston speed of 167 feet per minute, while the main engine made from 340 to 360 turns, with a piston speed of from 837 to 992 feet per minute.

2. The main engine is much more reliable, having a smaller number of moving parts.

3. A vacuum can be sustained independent of the main engine, facilitating its starting and handling, and increasing the manœuvering power of the boat.

4. In case of breakdown of the main engine, there need be no interruption of the firing, as the condenser and pumps are capable of dealing with all the steam the boiler can make.

For reasons given, this arrangement of independent engines needs no more supervision, and consequently demands no greater engine room force than is required when the pumps are worked off the main engine.

Auxiliary Engines.—The following are the auxiliary engines used:

1. A vertical compound engine secured to the bulkhead between engine and fire room, for running the fan blower. The blower takes its air through a large ventilator cowl on deck, and compresses it into the closed fire room. The air reaches the closed ash-pits through ducts built into the coal bunkers to

the right and left of the boiler. The ducts lead down from close under the deck, and are provided at their upper end with valves which close automatically when the pressure in the furnace and ash-pit exceeds that in the fire room, a thing which may happen with leaky boiler tubes. In this case, the steam backing down into the ash-pits and ducts, might escape into the fire room and scald the firemen. With these valves in operation, the steam must escape up the chimney. This arrangement proved satisfactory. The boiler was originally fitted with brass tubes (afterwards exchanged for iron ones); these frequently leaked without doing damage to the fire room force.

2. A horizontal, double-acting, auxiliary feed pump in the engine room. It is also a fire and bilge pump. A hand pump in the fire room can also be used for feeding the boiler.

3. Five ejectors, one in each of the larger compartments. These, together with the auxiliary feed pump, the engine room bilge pump, the circulating pump, and a portable hand pump on deck, have discharged on trial more than 400 tons of water from the boat in an hour, equivalent to five times its displacement. The pumping arrangements must, therefore, be characterized as ample.

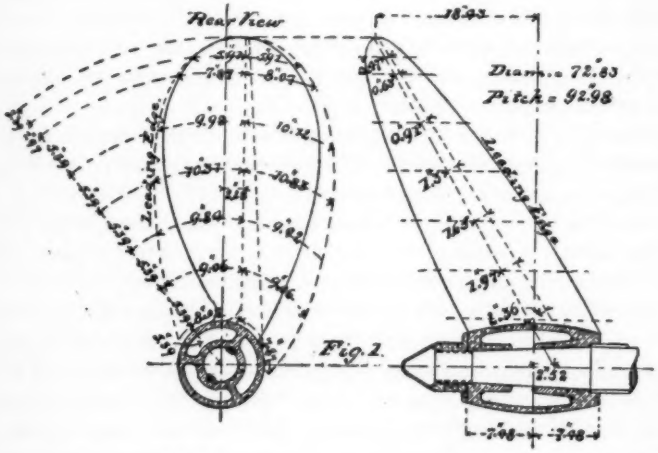
4. A Normandy distiller supplies drinking water and fresh water for the boiler to make up for waste. It is placed in the air lock between the engine and fire rooms.

5. A Kaselowski air compresser is located in the sixth compartment. The air for filling the torpedo reservoirs is compressed to between twelve hundred and thirteen hundred pounds. For discharging the torpedoes a pressure of from sixty-five to seventy pounds is used. By the side of the compressor is a coil of pipes serving as a reservoir for the high-pressure air.

6. In the forward conning tower is a steam and hand steering gear. By it both bow and stern rudder and the anchor windlass can be operated.

Screw.—The screw, represented in Fig. 1, is left-handed. It is three-bladed, of bronze, blades and hub cast together. The blades are bent aft like almost all torpedo boat screws in imitation of the Thornycroft screw. The inclination from the vertical is about

30°. The surface of the blades and hub is carefully smoothed by filing, to minimize friction. The screw is secured on the shaft by two keys driven lengthwise, and by a nut.



Besides the two torpedo tubes in the bow, the boats are armed with two 37 mm. Hotchkiss' rapid-firing guns placed on top of the conning towers.

The torpedo tubes are operated from the crew's compartment. At this part, the deck is arched more to give the necessary head-room. The four Schwartzkopff torpedoes, with which each boat is provided, are taken on board by means of small davits and sent into the torpedo room through the forward conning tower. Two are placed below the flooring in their charging ladles, while the remaining two are secured in a similar manner close under the deck ready for immediate use. When the torpedoes are to be discharged, the inboard shutters of the torpedo tubes are opened and the torpedoes shoved in. The inboard shutters are then closed, and the conical plugs closing the mouths of the tubes shoved open from the torpedo room. Meanwhile the reservoir shown just below the torpedo tube (Plate I) has been filled with air of from sixty-five to seventy pounds pressure. From the

forward conning tower this air may be admitted back of the torpedo, discharging it. While sliding through the tube, communication is established between the air reservoir in the torpedo and its Brotherhood propelling engine, so that on striking the water it becomes automobile.

Equipment.—Under this head are comprised compasses, means for transmitting orders, steering gear, anchor hoist, life saving apparatus, and the masting.

Care is taken to make the means of communicating orders as complete as possible. A telegraph and speaking tube connect each conning tower with the engine room. Similar communication exists between the engine and fire room. A speaking tube connects both conning towers.

There are two balanced rudders, one at the stern the other at the bow. The latter, when not in use, is raised up clear of the boat's bottom. It is lowered for use only when the boat is to make quick turns or to prevent leeway in rough weather. It is then operated by the steam steering gear in the forward conning tower. Ordinarily the stern rudder is the only one used. This rudder may be operated either by the steam gear, by a hand gear in the forward conning tower, by a special hand gear in the after tower, or by the hand wheel just forward of the latter.

The boats carry a Baxter anchor, which is not stowed on deck, but hauled up into the hawse hole, which is formed to receive it. Forward of the forward conning tower is a Baxter anchor hoist, which is operated either by hand or by the steering engine. The advantage of this anchor-hoisting gear is, that no one need expose himself to the danger of being washed overboard, the whole operation being controlled from the conning tower.

With the life-saving apparatus may be classed the pumping out machinery already described, the fire main, the life boat, life buoys and life preservers.

There is a special steam fire pump which can be used when steam is up. When no steam is up, a portable hand pump may be used.

A patent Francis lifeboat 12′ 6″ long is stowed above the fire room. Attached to the rail amidships are two life buoys. Life

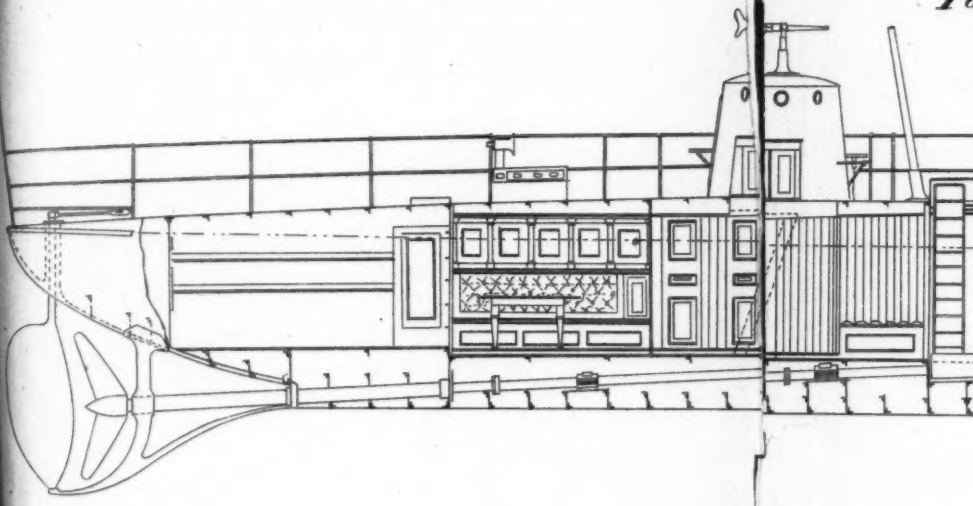
preservers are kept at the rail near the forward conning tower and in wake of the engine and fire room hatches. The boats have two light masts which can be unshipped, and two stay-sails.

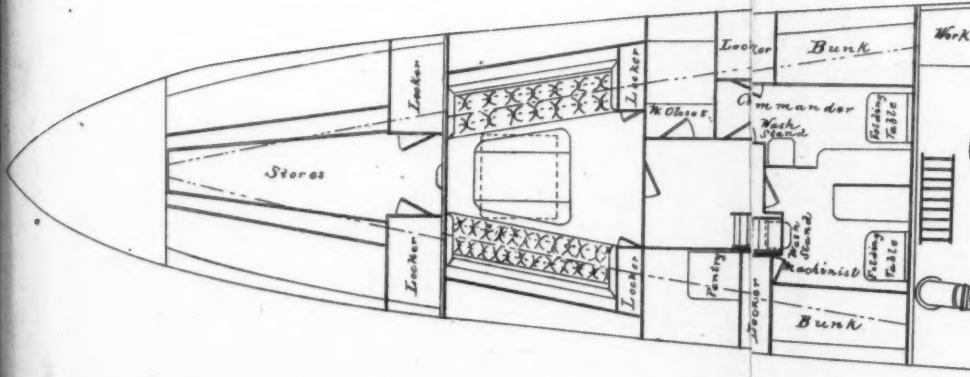
Trial trips.—The two boats already delivered passed their trials very successfully. The trials were measured mile runs over the two-mile course in the Bay of Eckernfoerde, and three hours' full power trials at sea. The speed on the measured mile, with an extra weight of six tons of coal on board, repeatedly reached 21.8 and 21.86 knots. The mean of six two-mile runs was 21.66, the contract requiring 21.5. As the boats had to make the turns at each end of the two-mile course with undiminished speed, and took nearly an hour to steam the six two-mile runs with the turns, these speeds are the actual speeds for the hour. On the three hours' run at sea an average speed of 21.3 knots was maintained, exceding the contract speed of 19.5 knots by 1.8. With the coal supply of twenty tons the boats can steam 2,100 knots at ten knots an hour.

The following are the results of a number of sets of indicator cards taken during the trials, the power varying from 217 I. H. P. to 1,233 (metric):

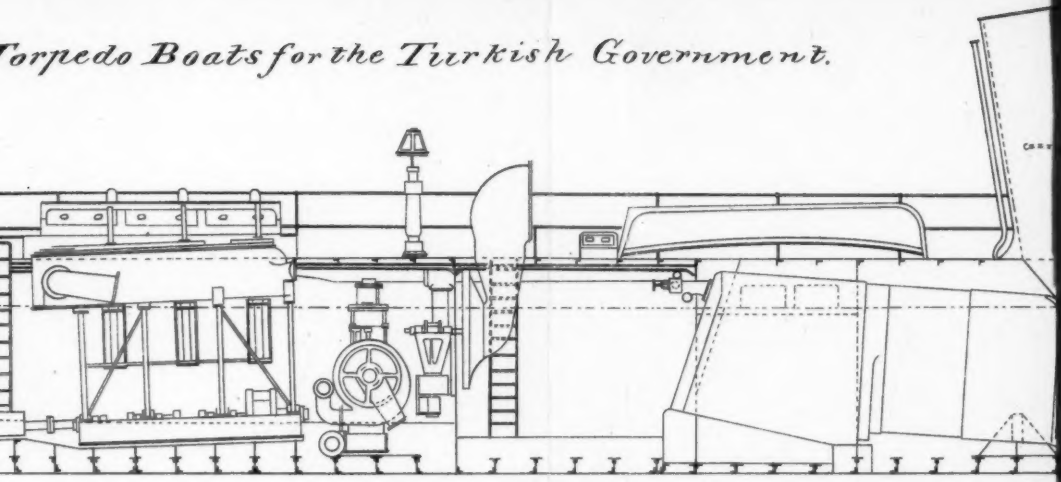
Number of revolutions, .	178	250	300	340
Boiler pressure, . .	169	159	173	179
Press. at H. P. cylinder, .	153	154	165	172
Vacuum, . . .	27 in.	25.6 in.	26 in.	26.5
I. H. P. in H. P. cylinder, .	38	199	249	471
I. H. P. in I. P. " .	95	154	295	363
I. H. P. in L. P. " .	84	148	313	399
Aggregate I. H. P., . .	217	501	857	1,233

Curves of performance.—In Fig. 2 are given the I. H. P. for observed speeds, together with the number of revolutions and the slip of the screw for the two boats already delivered. Unfortunately it was not always possible to obtain speed, revolutions and I. H. P., but the last two items could be and were obtained frequently. Moreover, as the displacement of both boats was at all times practically the same, and the pitch of



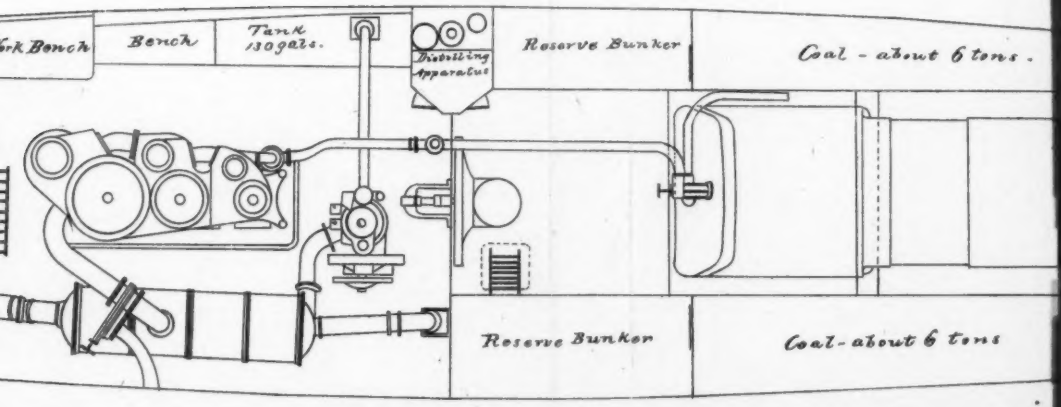

Locker
Locker
Locker
Locker
Stores
Locker
Bunk
Work
Commander
Wash Stand
Folding Table
Mess Stand
Folding Table
Machinist
Bunk

Torpedo Boats for the Turkish Government.



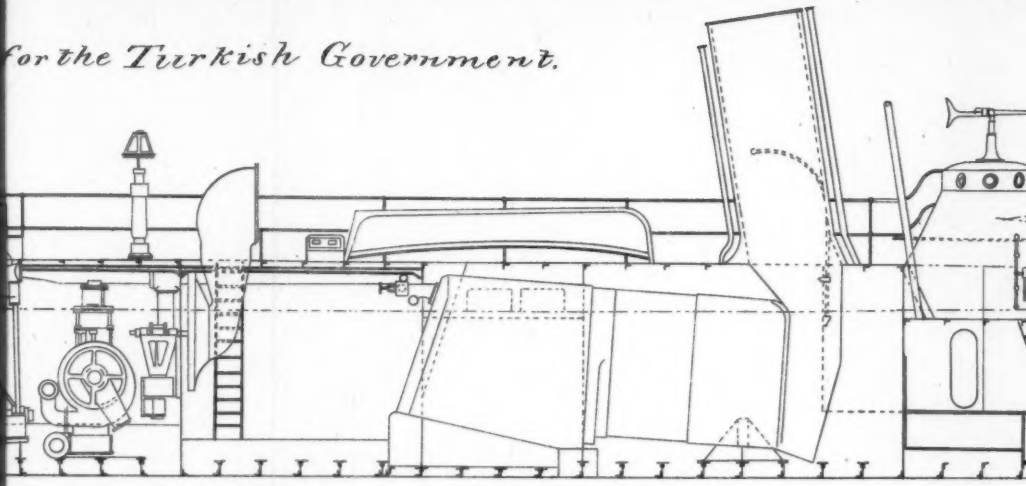
Longitudinal Section.

Scale 1:80



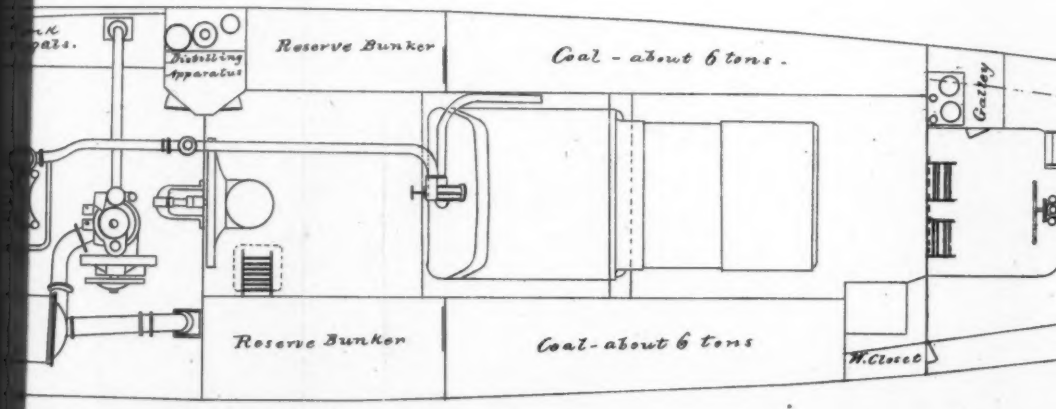
Plan.

for the Turkish Government.

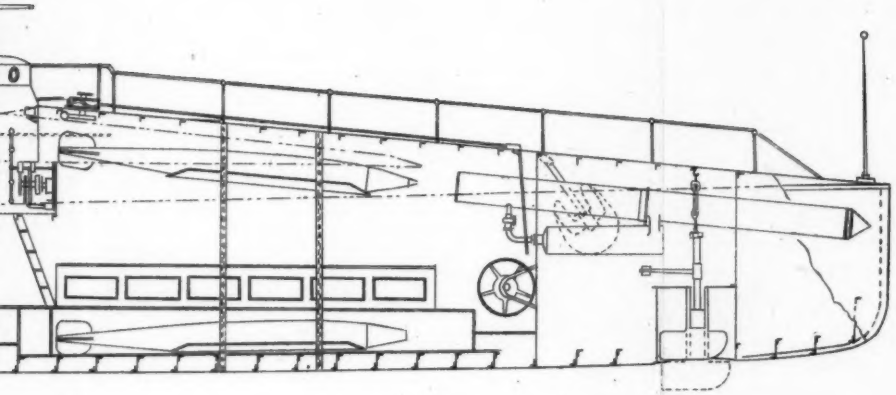


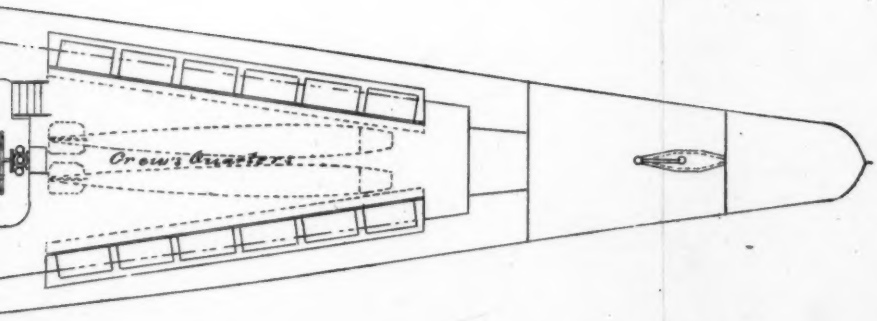
Longitudinal Section.

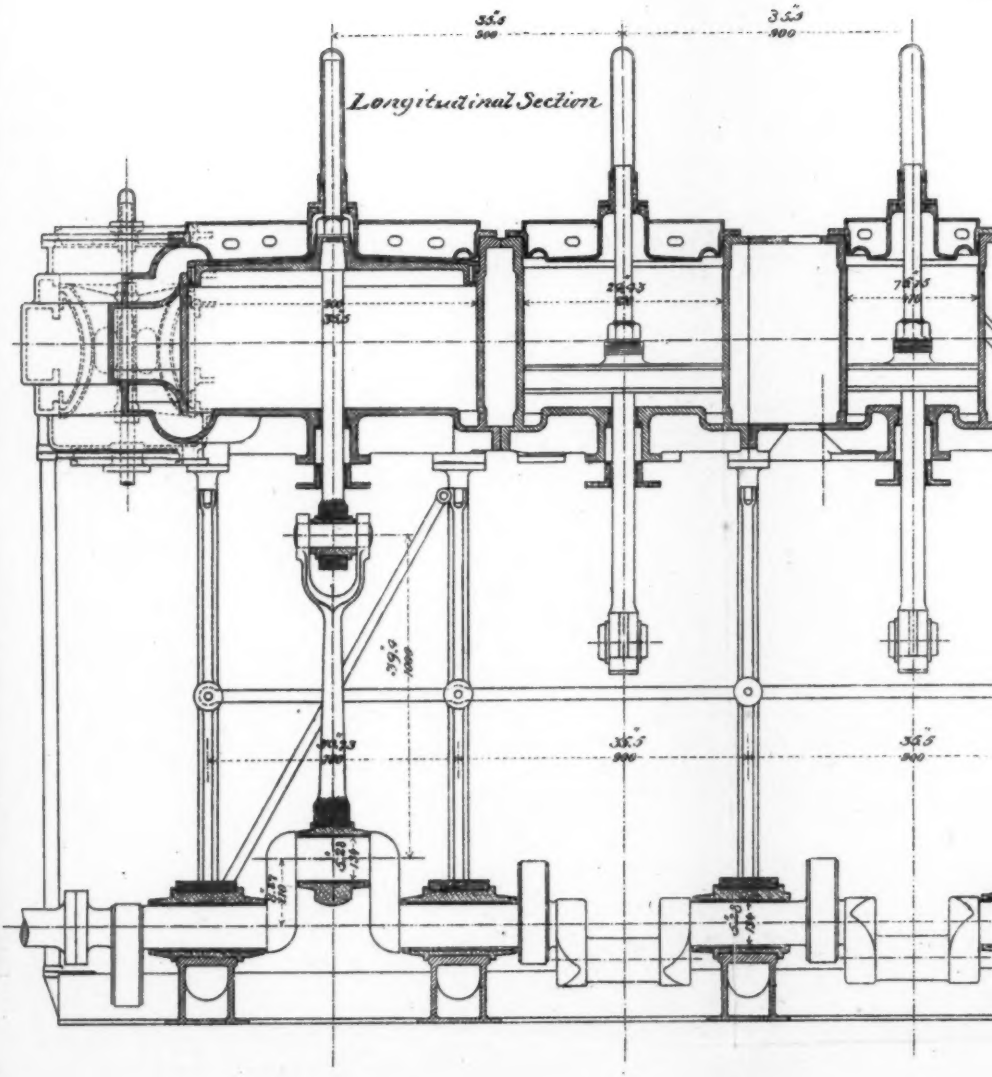
Scale 1:80

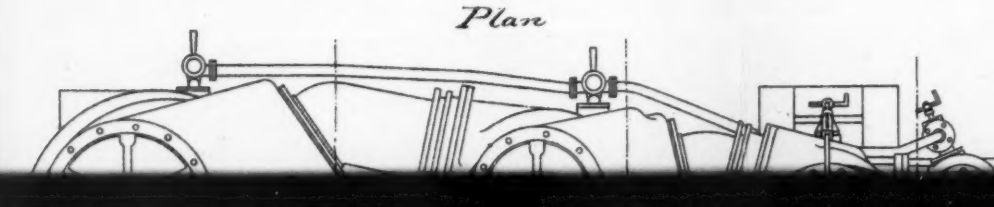


Plan.




Crew's Quarters

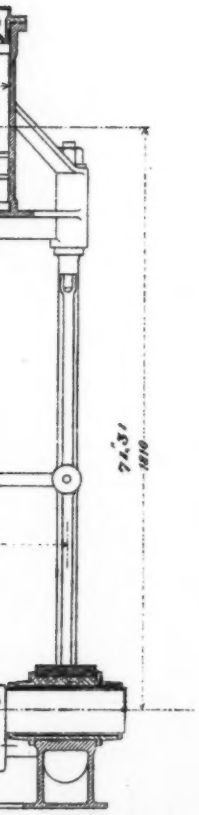

Longitudinal Section

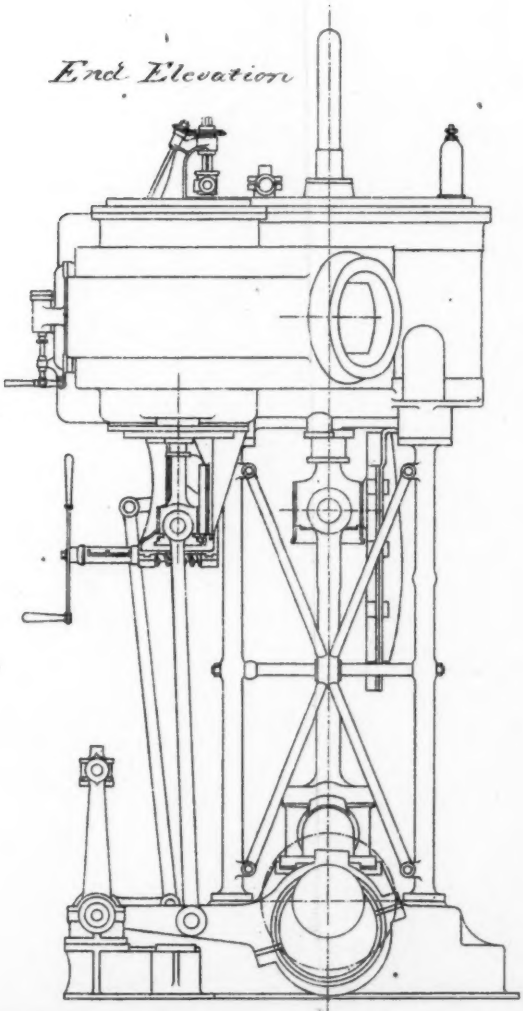

Plan

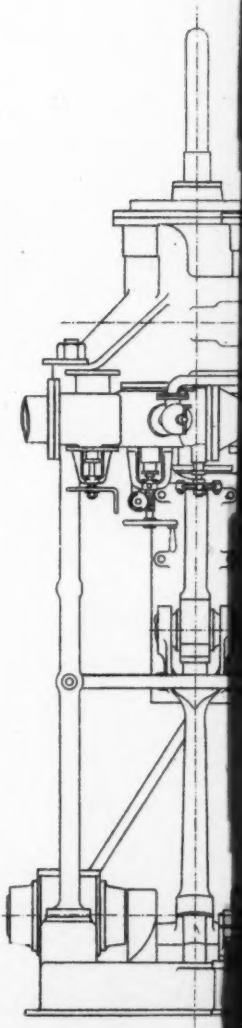
Torpedo Boats for the Turkish Government.

Engines & Boiler.

End Elevation







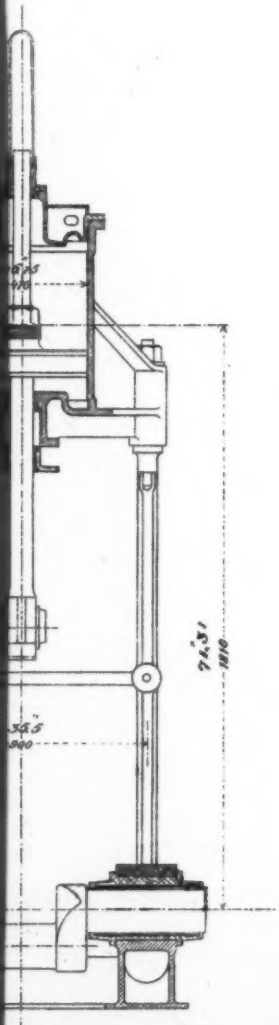
Scale of

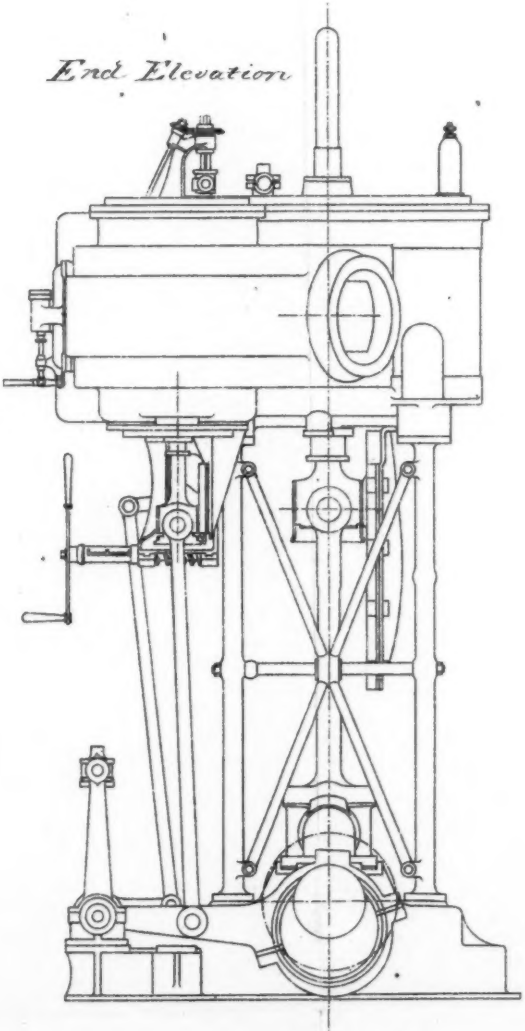
Heating surface - 160.78 sq ft.
Grate " " = 35.52 sq ft.
Heating surface / Grate = 4.3.94

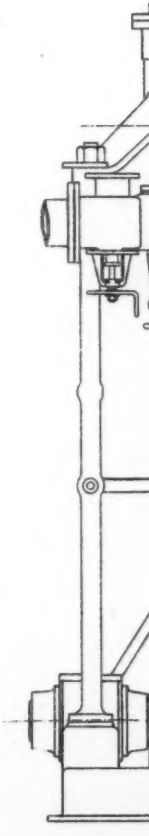
Torpedo Boats for the Turkish Government.

Engines & Boiler.

End Elevation







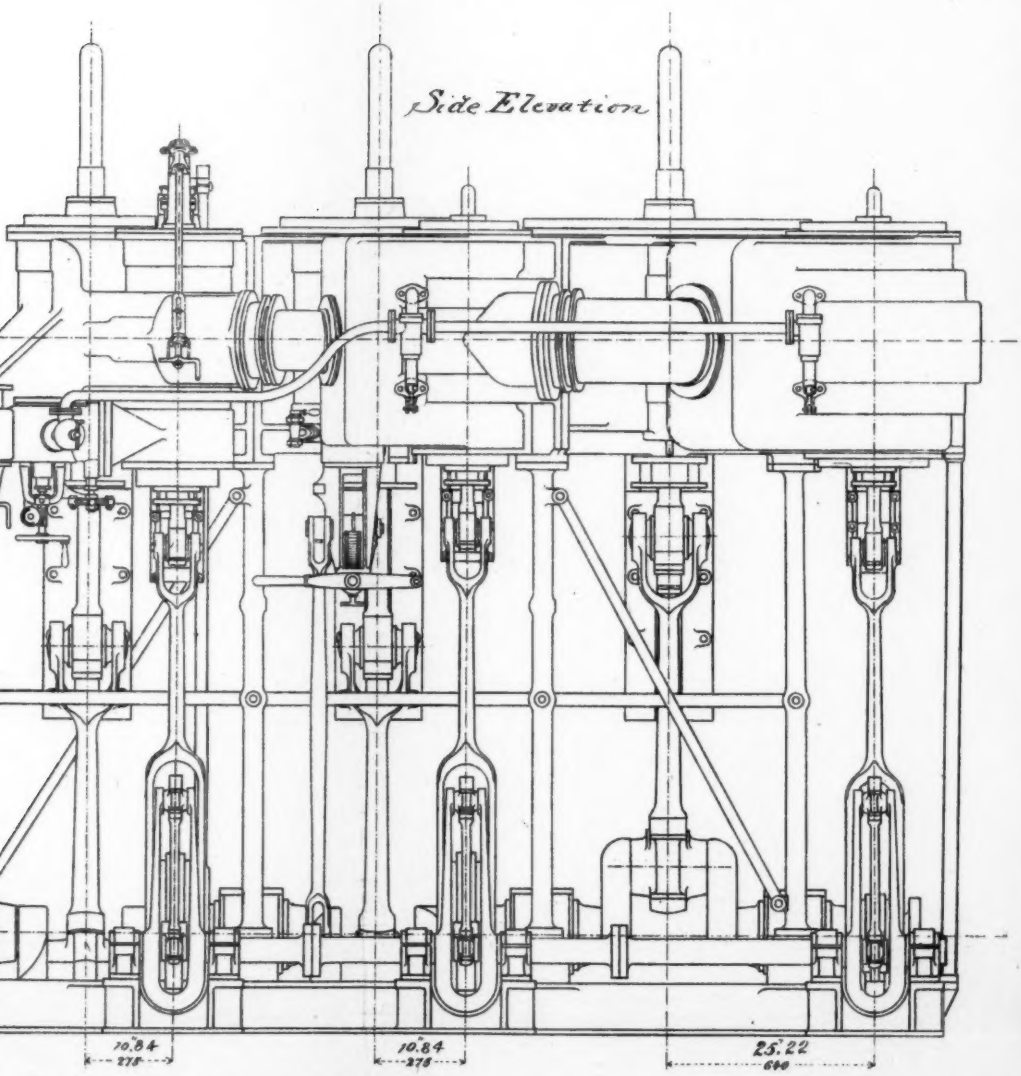
Heating surface - 1560.78 sq.ft.

Grate " " = 35.53 sq.ft.

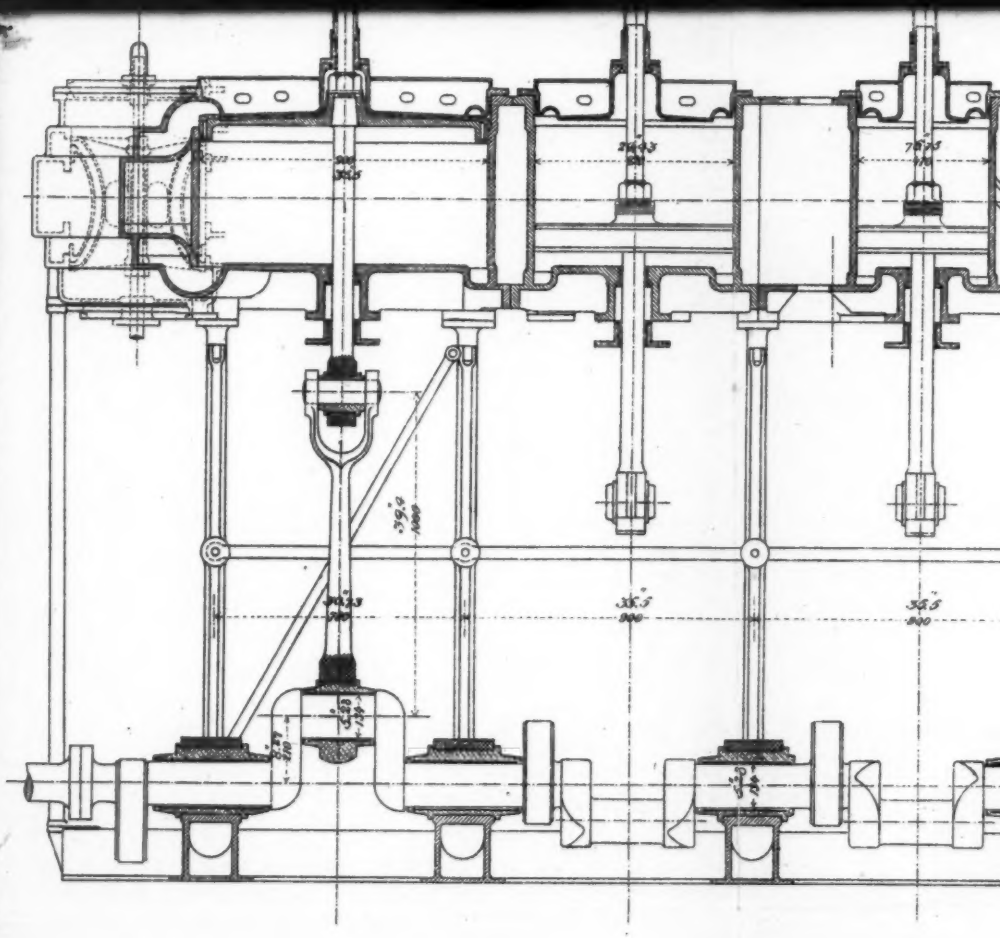
Heating surface

Plate II.

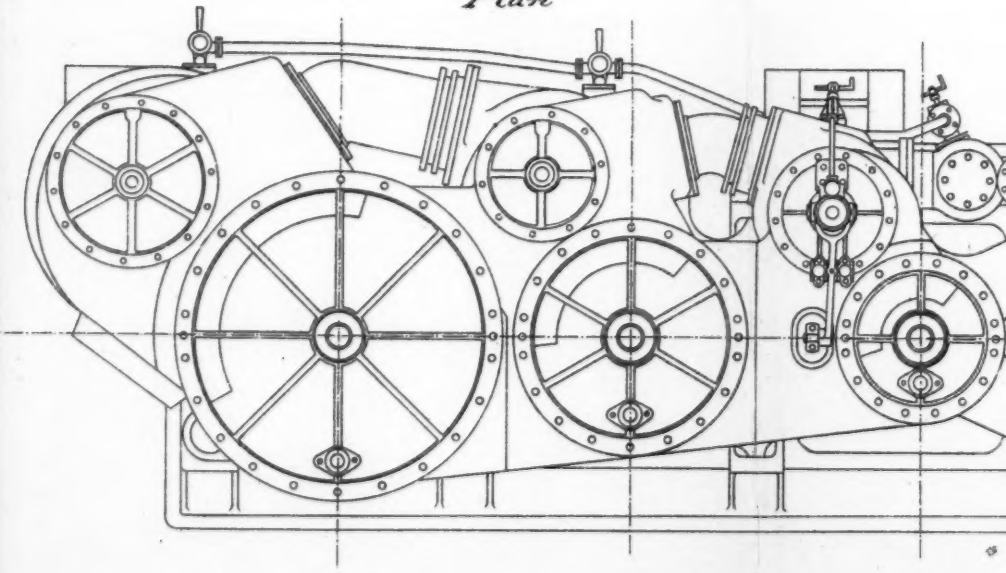
Side Elevation

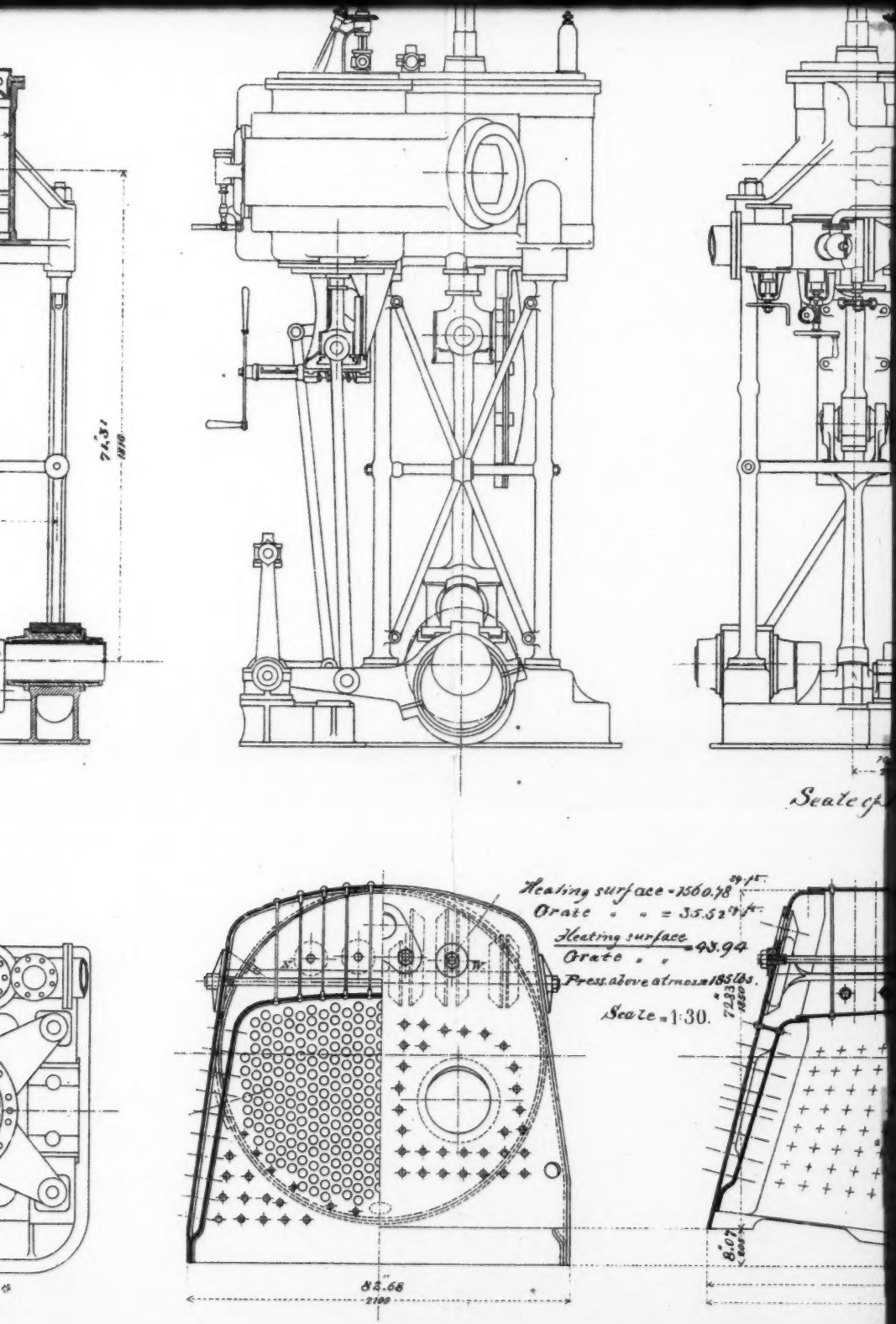


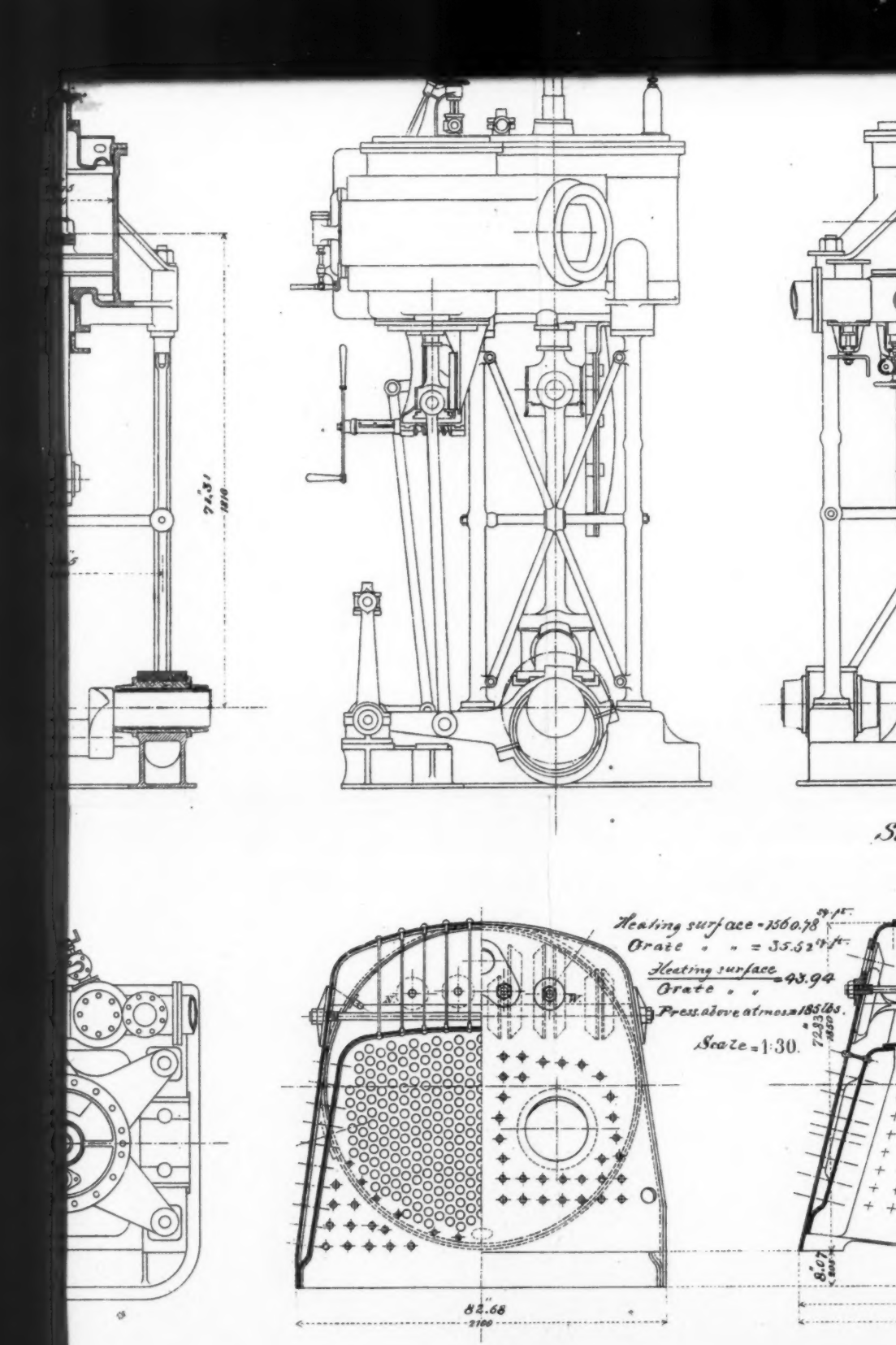
Scale of Engines - 1:20.

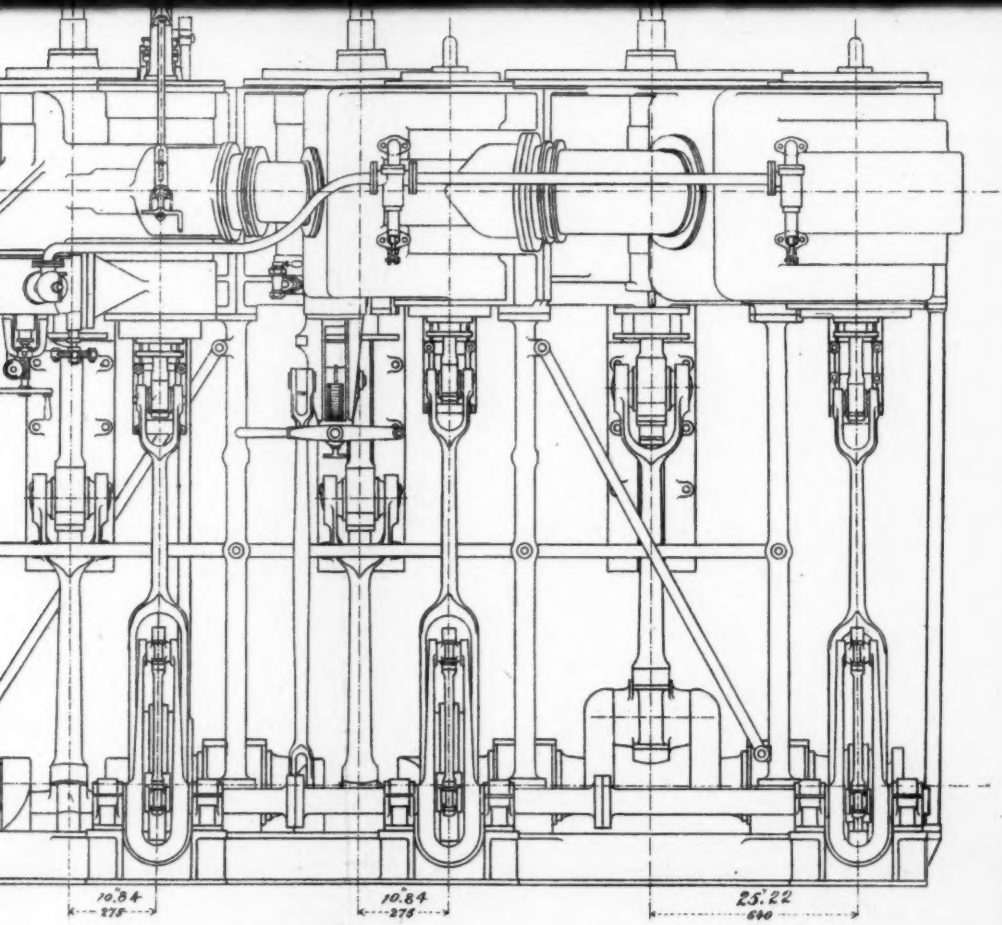


Plan

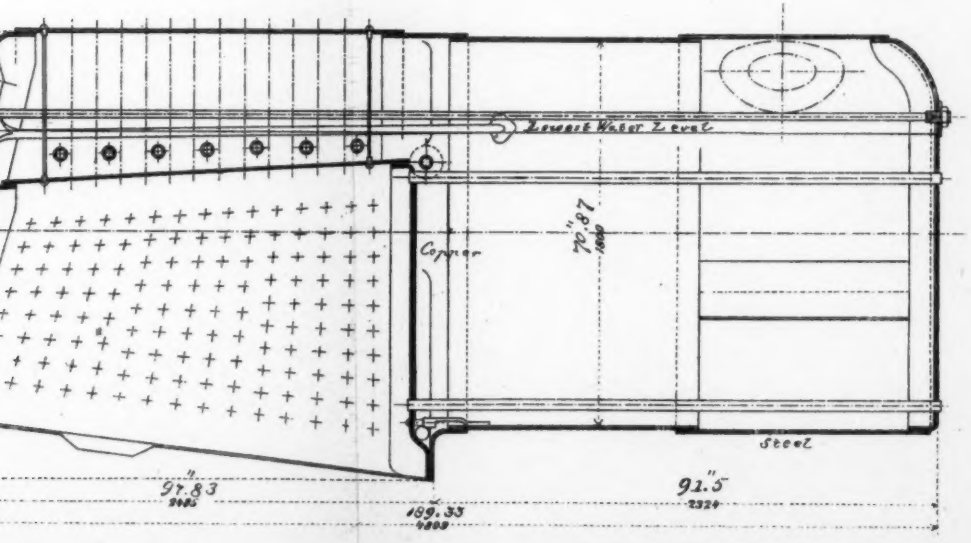



74.31
1890
Scale of
Heating surface = 1560.78 sq.ft.
Grate " " = 35.52 sq.ft.
Heating surface / Grate " " = 43.94
Press. above atmos. 185 lbs.
7233
1850
Scale = 1:30.
82.68
2100
8.07
205

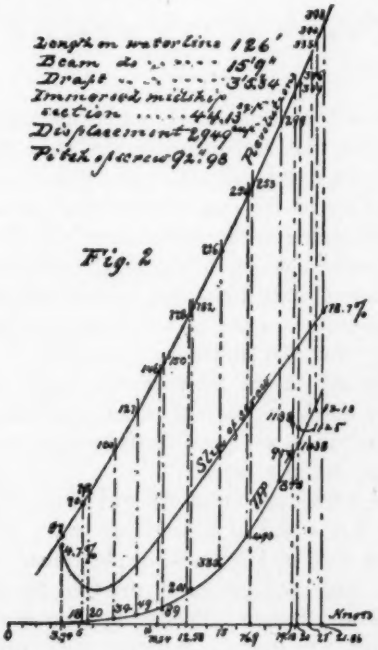

Heating surface = 1560.78 sq.ft.
Grate " " = 35.52 sq.ft.
Heating surface / Grate " " = 43.94
Press. above atmos. = 185 lbs.
Scale = 1:30.
24.31
810
82.68
2760
8.07
270
72.33
2410



Scale of Engines = 1:20.



both screws identical (= 7′ 8.″9), the missing data in regard to speed, revolutions and I. H. P. could be readily interpolated. In Fig. 2 these quantities are accordingly given for speeds ranging from 3.74 to 21.86 knots.



VI.

SECOND TRIAL U. S. S. BALTIMORE.

The contractors for this vessel, not being satisfied with the results of the first trial, and believing that they could do better on another, obtained the consent of the Navy Department, and the trial took place on the 15th of November last.

As previously stated in the JOURNAL (p. 347, Vol. I), the contractors made some changes in the machinery between the two trials, the most important being the increase of the pitch of the propellers to about 21.5 feet; an overhauling of the main valves to secure a freer exhaust and a later cut-off in H. P. cylinders; and an overhauling of the air pumps. On the trial a reserve supply of fresh water was carried in the double bottom to make up for losses and avoid any possible danger due to foaming from mixture of fresh and salt water.

On the second trial, the mean draught of the ship for the four hours was 18 feet 2 inches forward, 21 feet 7 inches aft, or a mean of 19 feet 10½ inches. The displacement was 4,500 tons. Everything having been prepared, the trial began at 8 A. M. and concluded at 12 noon. For the first hour and a half the wind varied from 3 to 5 on the Beaufort scale, with smooth and moderate sea, but for the remainder of the trial the wind's force was 7 and the sea rough; the wind was abeam except for the last quarter hour, when it was from ahead.

The performance of the machinery was admirable except the L. P. crank pin of port engine, which was for a time so hot that it seemed to threaten any further continuance of the trial, the stream of water from a deck hose which was run on it being thrown off so hot as to be almost scalding.

Pocahontas coal was used, as on the first trial. The two auxiliary boilers were used this time, thus increasing the grate sur-

face to 676 square feet, and the heating surface to 17,174.64 square feet.

(For complete data of *Baltimore's* machinery, see pp. 339–343 of Vol. I.)

SYNOPSIS OF STEAM TRIAL OF U. S. S. BALTIMORE, NOVEMBER 15, 1889.

(AVERAGES.)

		Starboard.	*Port.*
Steam at boilers		Gauges out of order.	
Steam at engines (per gauge)		125.1	123.44
Steam at 1st receiver (absolute)		62.2	59.28
Steam at 2d receiver (absolute)		20.4	21.75
Vacuum in condenser in inches of mercury		24.5	24.3
Revolutions of main engines per minute		116.42	116.08
Piston speed in feet per minute		814.94	812.56
Throttle, in eighths of full opening		8.	7.5
Air pressure in fire rooms in inches of water		2.09	2.08
Revolutions per minute of blowers		502.	630.
Temperatures in degrees Fahrenheit.	Fire rooms	93.	93.
	Engine rooms	70.	73.
	Injection	56.	56.
	Discharge	100.	97.
	Feed	129.	131.
Revolutions of circulating pumps per minute		177.	177.
Fraction of stroke completed when steam was cut off	H. P. cylinder	0.69	0.70
	I. P. cylinder	0.77	0.77
	L. P. cylinder	0.77	0.77
H. P. cylinder	Mean pressure	50.17	52.29
	I. H. P.	1,682.65	1,749.17
I. P. cylinder	Mean pressure	21.87	22.49
	I. H. P.	1,514.10	1,555.69
L. P. cylinder	Mean pressure	9.52	9.84
	I. H. P.	1,625.62	1,680.61
Aggregate equivalent mean pressure on L. P. piston		28.23	29.20
Collective I. H. P. each main engine		4,845.59	4,985.48
Collective I. H. P. both main engines		9,831.07	
I. H. P. of circulating pump engines		23.21	23.21
I. H. P. of blowers		110.86	
I. H. P. of feed pumps		28.81	
I. H. P. of all other auxiliaries		70.48	
Aggregate I. H. P. of all machinery		10,064.42	
Cubic feet per minute swept by L. P. piston per I. H. P.		8.45	8.19
Indicated thrust (I. H. P. of main engine only), pounds		63,890.	65,920.
Indicated thrust per square foot of helicoidal area of propellers, pounds		1,117.6	1,153.0

Indicated thrust per square inch of surface of thrust rings, pounds	61.2	63.15
*Speed of ship (mean for four hours), knots	20.096	
Slip of propeller, per cent	18.62	18.38
$\frac{S^2 \times \text{immersed midship section}}{\text{I. H. P.}}$	695.3	
$\frac{S^3 \times D^{\frac{2}{3}}}{\text{I. H. P.}}$	261.1	
†Coal used per hour, pounds	30,353.	
†Coal per hour per square foot of G. S., pounds	44.9	
†Coal per hour per I. H. P. (all machinery), pounds	3.15	
†Refuse per hour, pounds	2,124.	
†Refuse per hour, per cent	7.00	
The velocity of the steam through the ports when wide open was, in feet per second, for port engine—		
H. P. steam port	127.70	
I. P. steam port	150.10	
L. P. steam port	192.05	
H. P. exhaust port	104.47	
I. P. exhaust port	103.50	
L. P. exhaust port	138.60	
Main steam pipe	103.98	
Main exhaust pipe	132.02	
I. H. P. per square foot of G. S.	14.89	
Total H. S. ÷ I. H. P	1.707	
I. H. P. per ton of boilers, water and all fire room weights	20.51	
I. H. P. per ton of propelling machinery and water	10.53	
I. H. P. per ton of all machinery, spare gear, tools and water	10.12	
Cooling surface per I. H. P., square feet	1.256	
Wetted surface by Kirk's analysis, square feet	20,585.	
I. H. P. per 100 square feet of wetted surface, at 20.096 knots	48.89	
I. H. P. per 100 square feet of wetted surface, at 10 knots, reduced in ratio of 3.5 power	4.25	

MAXIMUM PERFORMANCE.

(Card No. 4, 8.45 to 9.00 A. M.)

	Starboard.	*Port.*
Steam at boilers (per gauge)	Gauges out of order.	
Steam at engines (per gauge)	130.5	126.0
Steam at 1st receiver (absolute)	64.7	64.0
Steam at 2d receiver (absolute)	21.4	24.0
Vacuum in condenser in inches of mercury	24.5	23.7
Throttle	Wide.	Wide.

*By patent log.
†Estimated.

Revolutions of main engines per minute	118.0	118.6
Piston speed in feet per minute	826.0	830.2
Mean pressure H. P. cylinder	51.64	52.14
Mean pressure I. P. cylinder	22.83	24.94
Mean pressure L. P. cylinder	9.87	10.20
Mean pressure aggregate equivalent on L. P.	29.26	30.54
I. H. P., H. P. cylinder	1,754.90	1,779.99
I. H. P., I. P. cylinder	1,601.88	1,759.62
I. H. P., L. P. cylinder	1,708.24	1,863.93
Collective I. H. P. each main engine	5,065.02	5,403.55
Collective I. H. P. both main engines	10,468.57	
I. H. P. circulating pumps	23.44	23.44
I. H. P. blowers	110.86	
I. H. P. feed pumps	28.81	
I. H. P. all other auxiliaries	70.48	
I. H. P. of all machinery	10,725.60	
Cubic feet per minute swept by L. P. piston per I. H. P.	7.83	7.37
Air pressure in fire rooms in inches of water	2.4	2.4
Cut-off in H. P. cylinder	0.69	0.75
Cut-off in I. P. cylinder	0.77	0.77
Cut-off in L. P. cylinder	0.77	0.77
I. H. P. per square foot of grate surface	15.49	
Total heating surface ÷ I. H. P.	1.638	
I. H. P. per ton of propelling machinery and water	10.95	
I. H. P. per ton of all machinery, spare gear, &c.	10.52	
Speed for card No. 4 (maximum I. H. P.)	19.57	
Slip of propeller per cent	21.83	20.21
$\frac{S^3 \times \text{immersed midship section}}{\text{I. H. P.}}$ for card No. 4	589.3	
$\frac{S^3 \times D^{\frac{2}{3}}}{\text{I. H. P.}}$ For card No. 4	194.5	
Maximum speed (card No. 12)	20.60	
Slip of propeller per cent	15.13	15.22
I. H. P. per card No. 12	9,568.47	
$\frac{S^3 \times \text{immersed midship section}}{\text{I. H. P.}}$ for card No. 12	786.5	
$\frac{S^3 \times D^{\frac{2}{3}}}{\text{I. H. P.}}$ for card No. 12	254.2	

SPEEDS, REVOLUTIONS AND SLIPS OF PROPELLERS.

Card No.	Beginning of 15 minute period.	Reading of the patent log.	Rate of speed in knots per hour for the 15 minutes.	Starboard.		Port.	
				Revolutions	Slip.	Revolutions	Slip.
1	8-00	13.80	19.98	117.07	19.57	117.3	18.94
2	15	18.65	19.98	117.00	19.51	117.4	19.78
3	30	23.50	19.98	118.47	20.51	118.6	20.58
4	45	28.35	19.57	118.00	21.83	118.6	20.21
5	9-00	33.10	19.77	117.07	20.41	117.5	20.70
6	15	37.90	19.77	118.67	21.48	118.1	21.48
7	30	42.70	19.77	116.80	20.23	117.3	19.96
8	45	47.50	19.98	118.00	20.19	117.3	19.92
9	10-00	52.35	20.39	117.07	17.91	116.7	17.65
10	15	57.30	20.39	115.47	16.77	113.1	15.00
11	30	62.25	20.19	115.60	17.68	114.1	16.60
12	45	67.15	20.60	114.40	15.13	114.6	15.22
13	11-00	72.15	20.39	114.27	15.90	114.8	16.26
14	15	77.10	20.19	114.80	17.11	114.2	16.67
15	30	82.00	20.19	113.60	16.23	114.1	16.60
16	45	86.90	20.39	116.40	17.44	115.1	16.46
	12-00	91.85					

An examination of the foregoing data shows at once that the increase in horse-power over the first trial was due to higher steam pressure, greater air pressure and the use of the two auxiliary boilers, coupled with better use of the steam due to freer passages. The speeds as given by the patent log show their unreliability, which is emphasized by the comparison of speeds, horse-powers and coefficients for cards 4 and 12. There can be no doubt that the *Baltimore* is a fast ship, and her trials on the measured mile will be awaited with great interest. Whether her bearings can be kept cool with the small number of oilers in her regular complement is open to doubt when it is noted that, with twelve in each engine room on the trial, the port low pressure crank pin heated so badly. As a result of this second trial, the contractors will receive the handsome premium of $106,442.

TORPEDO BOAT CUSHING.

The twin-screw torpedo boat *Cushing* was launched from the Herreshoff works at Bristol, R. I., on the 23d of January. In general appearance, she is similar to the well-known Thornycroft boats, but differs from them in internal arragement.

The following are her principal dimensions:

Length between perpendiculars,	137.5	feet.
Breadth, extreme,	15.05	"
Breadth at L. W. L.,	14.10	"
Draught, normal,	4.50	"
Displacement at above draught,	91.34	tons.
Tons per inch immersion,	3.02	"
Area of immersed midship section,	42.36	square feet.
Wetted surface,	1,701.00	" "
Deep load draught,	5.2	feet.
Wetted surface at same,	1,895.0	square feet.
Area of rudder,	14.0	"

She is built of steel, and is divided into eleven water-tight compartments. The frames are spaced 18 inches from center to to center, and the thickness of deck and bottom plating varies from $\frac{5}{64}$ to $\frac{1}{4}$ inch. The coal bunkers have a capacity of about 33 tons.

There are two sets of quadruple-expansion engines, placed in a common water-tight compartment, designed for a collective I. H. P. of 1,600, with a boiler pressure of 225 pounds per square inch. Each engine comprises five cylinders, each operating its own crank. The diameters are: H. P., 11¼; first I. P., 16; second I. P., 22½; two L. P., 22½ inches; the stroke of all being 15 inches. The steam valves are all of the piston type, operated from a counter shaft, worked by gearing from the crank shaft. The crank shafts, in two sections, are 5 inches in diameter, and the line and propeller shafts 5¼ inches.

There is one condenser, placed between the L. P. cylinders, all of the auxiliaries, as well as the L. P. cylinders, exhausting into it. All of the pumps are independent.

The original contract and specifications called for two Herreshoff boilers, but, at the request of the contractors, Thornycroft boilers were substituted. They are two in number, placed in separate water-tight compartments, one forward and one abaft the engine room, each supplying steam to one set of engines.

The screws are four-bladed, 50 inches in diameter.

The armament consists of three six-pdr. Hotchkiss rapid-fire guns and two Gatlings.

The contract requires a speed of twenty-three knots to be maintained for three consecutive hours. There is a premium of $2,000 for each quarter knot above twenty-four, and a penalty of $4,000 if the speed falls below twenty-two knots.

The contract price is $82,750.

THE ANNUAL MEETING of the Society, for the election of officers for the current year, will be held in Washington, on the 23d of April. Members who will not be able to attend are requested to authorize some one who will be present to vote for them.